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and Mem.Cncl.Inst.Metals, &c.;

AND

H. M. ROUNTHWAITE,
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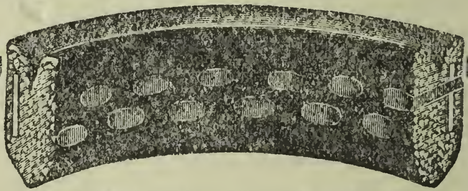

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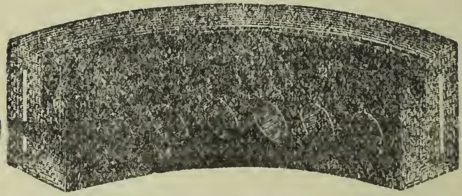

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# PREFACE TO FIRST EDITION.

A special Pocket-book of Memoranda, Tables, &c., has long been a desideratum with Marine Engineers. In the existing pocket-books, Marine Engineering matters are only dealt with generally, and such information as is given is in some cases very restricted, in others obsolete, and in all too scattered to be useful. We, ourselves, have experienced this want, and have heard on all hands the desire expressed for a Pocket-book in which Marine Engineering questions are dealt with thoroughly, are easy to find, and not "mixed up" with general information in such a way as to render the seeking of them difficult and tedious.

We therefore trust that in presenting this book to the public we have not only fulfilled the task we set ourselves, but have supplied this long-felt want in a manner that will prove satisfactory to all engaged in Marine Engineering affairs. While we have been careful to make the book of special value to Marine Engineers, we have omitted nothing, so far as we know, that would be of use and importance to others having to do with Ships and their machinery; at the same time, we have avoided the introduction of extraneous matter of only general interest, which would make the volume so bulky, and the arrangement of it so complex, as to very materially detract from its usefulness. Hence, we have, while not altogether neglecting past experience, but omitting information now almost only historic, devoted our attention generally to the most modern and approved practice.

We have dealt with steel as the material in general use, and not, as heretofore, an exceptional thing to be found only in high-class structures; the Tables of Weights, &c., are, therefore, given fully for this material.

Inasmuch as the practice in a considerable part of the Mercantile Marine is now more nearly approaching that followed in Naval ships, as to speed and economy of weight, than was formerly the case, the information and formulæ pertaining to light fast-running machinery have been elaborated and based on the most recent practice of the leading firms of Manufacturing Engineers.

In conclusion, we trust that the book may be received favourably, and found of use by practical men, and that any shortcomings may be overlooked on the score that it is the production of the spare moments of busy men, rather than of those having ample leisure.

A. E S.
H. M. R.

*October* 1893.

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# PREFACE TO THE FOURTEENTH EDITION.

Circumstances beyond the control of both Publishers and Author have caused the delay of the issue of this edition much longer than was anticipated when the preparations for it were commenced. Although this was unfortunate in a general way, the delay has permitted of the inclusion in this edition of much from the reports of the British Marine Engineering Design & Construction Committee, which was formed with the special object of bringing about the unification of the rules in use to govern the design and construction of Marine machinery which required a certificate of classification from any of the authorities controlling the merchant shipping of the United Kingdom, viz.: the Board of Trade, Lloyd's Register, the British Corporation Registry, and the Bureau Veritas.

This Committee consists of representatives of these four authorities and of the various technical and scientific institutions interested especially in Marine Engineering matters. Every branch of the profession has had able and active members on it to represent its side, so that every question has been carefully considered and concluded after examination from every point of view. Its reports on Boilers and Shafting have been made public, and, while the four authorities have not thought fit to adopt the whole of the recommendations contained in them, a code has been drawn up based on and largely composed of them, which all of these authorities will embody in their several books of Rules and Regulations as issued for the guidance of the Surveyors, etc., together with the domestic arrangements special to each of them.

This means that now any design will be accepted and passed for the construction of a boiler and have the same working pressure allowed, whether presented to the Board of Trade, to Lloyd's Register, or to the other Register Societies for classification.

The same treatment will be also accorded to the proposals or designs for shafting and other important items of ship machinery installations.

Consequent on all this, many of the tables in this book have had to be changed, and some others added, so that generally in these and some other respects this edition is superior to the previous ones.

A Committee of the British Engineering Standards Association is engaged in the work of standardising engine and boiler details, whereby economy in manufacture and convenience in carrying on the construction and upkeep of the machinery on shipboard may be effected, and that with no loss of efficiency.

A. E. SEATON.

Westminster, *November* 1919.

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# MARINE ENGINEERING RULES AND TABLES.

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## MARINE ENGINES—VARIOUS KINDS OF.

**The prime movers employed on shipboard** for driving the propellers are to-day:—

(1) Reciprocating Engines; (2) Turbines; (3) Combinations of both.

1. **Reciprocating Engines** are worked by means of steam or the combustion within their cylinders of gas or spray from liquid fuels, or the gas from solid fuel.

*Steam-driven reciprocators* are direct acting, inverted in the case of screw ships and inclined for paddle ships. They are invariably of one of the compound types—that is, the steam is expanded and acts in a series of cylinders, instead of in one only. Marine engines of all steamers, except very special ones, such as tugs, where economy of fuel is not of prime importance, are of the triple or quadruple expansion principle, so that there are three or more cylinders to each. Fast-running engines in express steamers and naval ships, whether triple or quadruple, have usually four cylinders operating on four cranks.

*Oil- or gas-driven reciprocators* have usually four cylinders at least, each operating on its own crank. The two-stroke cycle, whereby an explosion takes place on alternate strokes in a single-acting cylinder, is the common practice. The double-acting oil engine is not yet accepted as satisfactory.

Except for small craft, heavy oil is the only liquid fuel, and the Diesel system, wherein the air alone is compressed highly—usually to 35 atmospheres,—after which the oil is sprayed in and ignited by contact with the hot compressed air, is the one generally accepted and used. Installations of producer-gas plants, whereby engines are supplied with gaseous fuel, are being tried, but they do not commend themselves, as against oil engines, from the greater weight and space occupied by them and the producers.

*Semi-Diesel engines* working with less compression of air alone, but having hot bulb or plate igniters, are now in general use.

2. **The Turbine** is a rotatory engine and essentially a velocity machine, as against the previous kind, which are all actuated by pressure. It derives its motion and power entirely from the kinetic energy of the steam particles, to which great velocity is imparted by the expansion during the lowering of pressure from the initial to that at exit to the condenser.

The expansion may take place in one stage, as in the De Laval turbine, or it may be in a series of stages, just as in compound reciprocators.

*Other rotatory machines* whose motion and power are due to steam pressure acting on pistons or their equivalents have been tried, but in small units only have they been successful.

3. **The combination** of a turbine with a reciprocator has been carried out on a very large scale in R.M.S. *Olympic*, and on other ocean-going steamships of various sizes, all with satisfactory results. By means of the low-pressure turbine the energy remaining in the exhaust steam from a triple compound reciprocator is fully utilised, and the very high vacuum so easily and cheaply maintained in a marine condenser thus made full use of effectively. The saving in fuel by this combination is generally about 15 per cent. over that of the triple or quadruple engine, or there is an increase in power developed to a corresponding extent from the same expenditure of fuel in each case.

*Reciprocators* are nearly always connected direct to the propeller shafting, as their rate of revolution is not unsuitable to that of the propellers.

*Turbines* are now often geared by pinion and wheels to the propeller shaft, so that they may not have so low a rate of revolution as that required for the high efficiency of a satisfactory screw. The efficiency of the turbine to be good requires high peripheral velocity or a large number of stages for expansion, the former being desirable.

*Ungeared Turbines* are usually divided, so that in small ships there are three screws, and in large full-powered ones four screws. Of late years, however, the twin-screw arrangement, with a complete turbine to each, has been preferred.

*Geared Turbines* may be each divided into two parts, the high pressure and the low pressure. For cargo ships of low power one screw is usual, for higher powers multiple screws obtain, so that naval ships of very high speed have as many as four screws. The gearing may be single for small powers, but now is usually double for larger powers, and is preferable.

*Large ships with high power*, even when driven by reciprocators only, are often supplied with three screws, especially those of foreign countries, and there are definite advantages claimed for the system.

## ENGINE POWER—MEASUREMENT OF.

**Nominal Horse-power**, as understood by Watt, was a measure of the commercial value of an engine, being the power it might be expected to develop in ordinary work. Gradually, however, as it became possible to construct boilers to supply steam at higher than atmospheric pressure, the powers developed by engines exceeded the nominal horse-powers, until Watt's rule ceased to have this meaning and value, and the general use of the indicator and other means of ascertaining the actual power developed by the engine when working have caused the decline in the use of the expression. But there remains the need for some means of expressing in simple figures the size of a reciprocating engine for purposes of comparison and commerce. There are also other and technical demands for such a denomination, as will be shown.

Thus, for many years, a marine engine was expected to indicate about five times its nominal horse-power. The rule then in use was as follows:—

$$\text{N.H.P.} = \frac{\text{Sum of squares of piston diameters}}{\text{30 to 33 (according to district)}},$$

boiler pressure, piston speed, &c., being left entirely out of consideration. It is, of course, hardly necessary to say that such a rule was quite useless for any scientific purpose; whilst, even for commercial purposes, it gave only a very imperfect idea of the relative values of different engines. Its use, however, still survives in some districts, the divisor being 30, the normal stroke ·618 of diameter of L.P. cylinder, and the normal heating surface 16 square feet per N.H.P.

The matter was still in this chaotic condition when, in 1883, Mr Seaton, in his "Manual of Marine Engineering," suggested the use of E.H.P., or **Estimated Horse-power**, which is now calculated in the following way: —

**Rule 1.** *Seaton's.* $\text{E.H.P.} = \dfrac{D^2 \times \sqrt{P} \times S \times R}{Z}$

D is the diameter of L.P. cylinder and S the stroke of piston, both in inches; P is the absolute boiler pressure; R, the revolutions per minute.

For naval ships with overloads, $Z = 85{,}000$.
,, short passage express steamers, $Z = 91{,}000$.
,, long ,, ,, $Z = 94{,}500$.
,, passenger cargo steamers, $Z = 97{,}000$.
,, cargo steamers, $Z = 105{,}000$.

**Rule 1a.** *Seaton's.* $\text{E.H.P.} = \dfrac{D^2 \times WP \times S \times R}{(r+2) \times 140{,}000}$

WP is the working or boiler pressure.
$r$ is the ratio of the L.P. to the H.P. cylinder capacity.

These formulæ give a very close approximation to the horse-power actually indicated when working at full speed.

In 1888, the North-East Coast Institution of Engineers and Shipbuilders proposed the following very complete formulæ :—

**Rule 2.** *N.E.C. Institution.* $\text{N.I.H.P.} = \dfrac{(D^2\sqrt[3]{S} + 3H\sqrt[3]{P})}{100}$

Where N.I.H.P. = Maximum normal indicated horse-power, on loaded trial trip, of surface-condensing screw engines, working at any pressure between 50 and 250 lbs., under "normal" conditions.
$D$ = Diameter of L.P. cylinder, in inches (if more than one, $D^2$ must equal sum of squares).
$S$ = Stroke, in inches.
$P$ = Working pressure, in lbs., above atmosphere.
$H$ = Heating surface of boilers in sq. feet.
$P_m$ = Mean pressure, in lbs., referred to L.P. cylinder.

The conditions assumed as "normal" are as follows:—That

(1) Steam of all pressures is expanded down to the same terminal pressure;

(2) Expansion is effected with same degree of efficiency for all pressures;

(3) Piston speeds are proportional to cube roots of strokes, and, further, actual loaded trial-trip piston speed may be taken as $144\sqrt[3]{S}$;

(4) In all cases where relative proportions of engine and boiler prevent (1) being fulfilled without violating (3), the coal consumption will not be affected, but will be constant for the same boiler pressure;

(5) Boilers are of usual proportions and construction, and the horse-power proportional to heating surface (H), and to cube root of pressure ($\sqrt[3]{P}$); and further, actual loaded trial-trip horse-power may be taken as $\frac{H\sqrt[3]{P}}{16}$;

(6) Efficiency of engine mechanism is constant, and the propeller such that engines will utilize boiler power, referred to in (5), in the manner prescribed in (3) and (4).

As a result of (1) and (2), it follows that mean pressure referred to L.P. cylinder ($P_m$) may be assumed as proportional to cube root of boiler pressure ($\sqrt[3]{P}$), and further, that its actual loaded trial-trip value may be taken, without sensible error, as $5{\cdot}6\sqrt[3]{P}$.

**The normal relation between engines and boilers** is expressed by the equation.

$$H = \frac{D^2\sqrt[3]{S}}{3{\cdot}25}.$$

The results obtained by the Rule 2 for N.I.H.P., if divided by 6, give quantities very near those found by the old nominal horse-power rule.

It is also claimed that,—for machinery of the same type and design, proportions and arrangement, built of similar materials, under similar circumstances, to similar factors of safety, and not differing very widely in size,—the weights and costs will vary almost exactly as N.I.H.P.

**For paddle engines** the same rule may be used, with the co-efficients altered to suit the piston speeds usual for these engines. Assuming that under (4), piston speed of paddle engines may be taken at $90\sqrt[3]{S}$, the rule may, without sensible error, be written,—

**Rule 3.** *Paddle Engine.* $\text{N.I.H.P.} = \dfrac{(D^2\sqrt[3]{S} + 5H)\sqrt[3]{P}}{160}$

and the normal relation between engines and boilers will be expressed by,—

$$H = \frac{D^2\sqrt[3]{S}}{5{\cdot}2}.$$

The "Standard practice" formerly obtaining on the North-East Coast of England for triple compound engines working with steam of 160 lbs. pressure was as in Table I. To-day N.H.P. is not used there; in its place an estimated I.H.P. is taken for cargo steamer engines designed on Standard Rules (*vide* Appendix L) for a working pressure of 180 lbs. This I.H.P. is $= D^2 \times S \times N \div 700$.

Where D is the diameter of the L.P. cylinder in inches, S is the stroke in feet, and N the revolutions per minute, which by rule $= 32(S + 4) \div S$.

Taking the stroke in inches as S, and combining these, then estimated I.H.P. $= D^2(S_1 + 4) \div 262{\cdot}5$.

**Table I.—Sizes of Cylinders, &c., and corresponding N.H.P.**

| N.H.P. | Diars. of cyls. in inches. | | | Stroke in ins. | N.H.P. | Diars. of cyls. in inches. | | | Stroke in ins. |
|---|---|---|---|---|---|---|---|---|---|
| | H.P. | M.P. | L.P. | | | H.P. | M.P. | L.P. | |
| 20 | 8 | 13 | 22 | 18 | 140 | 19 | 31½ | 51½ | 36 |
| 30 | 9½ | 15½ | 26 | 21 | 150 | 20 | 32½ | 53 | 36 |
| 40 | 11 | 18 | 30 | 21 | 160 | 20 | 33 | 54 | 39 |
| 50 | 12½ | 20 | 33 | 24 | 170 | 20½ | 34 | 56 | 39 |
| 60 | 13½ | 22 | 36 | 24 | 180 | 21 | 35 | 57 | 39 |
| 70 | 14 | 23 | 38 | 27 | 190 | 21½ | 36 | 59 | 39 |
| 80 | 15½ | 25 | 40 | 27 | 200 | 22 | 36½ | 60 | 42 |
| 90 | 16 | 26 | 43 | 30 | 225 | 23 | 38 | 63 | 42 |
| 100 | 16½ | 27 | 45 | 30 | 250 | 24½ | 40 | 66 | 45 |
| 110 | 17 | 28 | 46 | 33 | 275 | 25 | 41½ | 69 | 45 |
| 120 | 18 | 29 | 48 | 33 | 300 | 26 | 43 | 71 | 48 |
| 130 | 18½ | 30 | 50 | 33 | ... | ... | ... | ... | ... |

**Rule 5.** *Board of Trade Rule* for registration purposes:—

$$\text{N.H.P.} = \frac{(3H + D^2\sqrt[3]{S})\sqrt[3]{P}}{700}$$

where H = heating surface of main boilers in square feet.
D² = square of diameter of low-pressure cylinder, or sum of squares of diameters of cylinders in non-compound engines, measured in inches.
S = length of stroke of engines in inches.
P = pressure of main boilers.

**Rule 6.** *Lloyd's Rule* for determining amount of survey fees, &c., is as follows:—

$$\text{N.H.P.}=\frac{P+C}{K}\left(\frac{D^2\sqrt{S}}{100}+\frac{H}{15}\right)$$

where D is diameter of L.P. cylinder in inches; S, stroke in inches; H, heating surface in sq. feet; P, working pressure in lbs. per sq. inch; and

$$C=\begin{cases}340 & \text{where boiler pressure is below 160 lbs.}\\ 590 & \text{,, \quad ,, \quad 160 lbs. or above.}\end{cases}$$

$$K=\begin{cases}1000 & \text{,, \quad ,, \quad below 160 lbs.}\\ 1500 & \text{,, \quad ,, \quad 160 lbs. or above.}\end{cases}$$

If the boilers are fitted for other than natural draught, $\frac{H}{12}$ is to be substituted for $\frac{H}{15}$. H includes surfaces of tubes, of back tube-plates, and of furnaces and combustion chambers down to level of fire bars.

Since the capacity of the L.P. cylinder is the true measure of the size of the engine, and as the other cylinders have a relation to the L.P. which does not vary largely in any but naval engines; and, further, as the standard stroke in general practice varies with the diameter of cylinders, the following gives a good criterion of the size of the engine, which may be usefully employed for various purposes:—

**Rule 7.** *Seaton's* (2). $\text{N.H.P.}=D\times S\div X$,

where D and S are as before the diameter of L.P. cylinder and stroke in inches, and X a factor which for compound engines is 15, for triples 12·5, and for quadruples 10·5, and extension of this is—

**Rule 8.** *Seaton's* (3). $\text{N.H.P.}=\frac{D\times S\times\sqrt{P}}{150}$ for all engines

where P is the absolute boiler pressure.

## Table II.—Nominal Horse-power of Triple Compound Engines (D × S ÷ 12·6).

| Diameter of Low-Press. Cylr. | Strokes in inches. | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 21 | 24 | 27 | 30 | 33 | 36 | 39 | 42 | 45 | 48 | 51 | 54 | 57 | 60 | 63 | 66 | 69 | 72 |
| ins. 30 | 50 | 56 | 64 | 70 | 77 | 84 | 91 | 98 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| 32 | 53 | 61 | 68 | 76 | 84 | 92 | 100 | 107 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| 34 | 56 | 64 | 72 | 81 | 89 | 97 | 105 | 113 | 121 | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| 36 | 60 | 68 | 77 | 85 | 94 | 102 | 111 | 120 | 128 | 137 | .. | .. | .. | .. | .. | .. | .. | .. |
| 38 | 63 | 71 | 80 | 89 | 98 | 107 | 116 | 126 | 135 | 144 | 153 | .. | .. | .. | .. | .. | .. | .. |
| 40 | 66 | 76 | 85 | 95 | 104 | 114 | 123 | 133 | 142 | 152 | 161 | 171 | .. | .. | .. | .. | .. | .. |
| 42 | .. | 80 | 90 | 100 | 110 | 120 | 130 | 140 | 150 | 160 | 170 | 180 | 190 | .. | .. | .. | .. | .. |
| 44 | .. | 83 | 94 | 104 | 115 | 125 | 136 | 146 | 157 | 167 | 178 | 188 | 198 | 209 | .. | .. | .. | .. |
| 46 | .. | 87 | 98 | 109 | 120 | 131 | 142 | 153 | 164 | 175 | 186 | 197 | 208 | 219 | .. | .. | .. | .. |
| 48 | .. | .. | 102 | 114 | 125 | 136 | 148 | 159 | 170 | 182 | 193 | 205 | 216 | 228 | .. | .. | .. | .. |
| 50 | .. | .. | 107 | 119 | 130 | 142 | 154 | 166 | 178 | 190 | 202 | 214 | 226 | 238 | .. | .. | .. | .. |
| 52 | .. | .. | .. | 123 | 136 | 148 | 161 | 173 | 185 | 198 | 210 | 222 | 235 | 247 | .. | .. | .. | .. |
| 54 | .. | .. | .. | 128 | 141 | 154 | 167 | 180 | 192 | 205 | 218 | 231 | 244 | 257 | .. | .. | .. | .. |
| 56 | .. | .. | .. | 133 | 146 | 160 | 173 | 186 | 200 | 213 | 226 | 240 | 253 | 266 | .. | .. | .. | .. |
| 58 | .. | .. | .. | .. | 151 | 165 | 179 | 193 | 207 | 221 | 234 | 248 | 262 | 276 | .. | .. | .. | .. |
| 60 | .. | .. | .. | .. | 157 | 171 | 185 | 200 | 214 | 228 | 242 | 257 | 271 | 285 | .. | .. | .. | .. |
| 62 | .. | .. | .. | .. | .. | 177 | 191 | 206 | 221 | 236 | 250 | 265 | 280 | 295 | .. | .. | .. | .. |
| 64 | .. | .. | .. | .. | .. | 182 | 198 | 213 | 228 | 243 | 259 | 274 | 289 | 304 | 320 | .. | .. | .. |
| 66 | .. | .. | .. | .. | .. | 188 | 204 | 220 | 235 | 251 | 267 | 282 | 298 | 314 | 330 | .. | .. | .. |
| 68 | .. | .. | .. | .. | .. | .. | 210 | 226 | 242 | 258 | 274 | 290 | 306 | 323 | 339 | .. | .. | .. |
| 70 | .. | .. | .. | .. | .. | .. | 216 | 233 | 250 | 266 | 283 | 300 | 316 | 333 | 350 | .. | .. | .. |
| 72 | .. | .. | .. | .. | .. | .. | 222 | 240 | 257 | 274 | 291 | 308 | 325 | 342 | 360 | 377 | .. | .. |
| 74 | .. | .. | .. | .. | .. | .. | .. | 246 | 264 | 282 | 299 | 317 | 334 | 352 | 370 | 387 | .. | .. |
| 76 | .. | .. | .. | .. | .. | .. | .. | 253 | 271 | 289 | 307 | 325 | 343 | 362 | 381 | 399 | .. | .. |
| 78 | .. | .. | .. | .. | .. | .. | .. | 260 | 278 | 297 | 315 | 334 | 352 | 371 | 390 | 408 | .. | .. |
| 80 | .. | .. | .. | .. | .. | .. | .. | 266 | 285 | 304 | 323 | 342 | 361 | 380 | 400 | 419 | 439 | .. |
| 82 | .. | .. | .. | .. | .. | .. | .. | .. | 292 | 312 | 331 | 351 | 371 | 390 | 410 | 429 | 449 | .. |
| 84 | .. | .. | .. | .. | .. | .. | .. | .. | 300 | 320 | 340 | 360 | 380 | 400 | 420 | 440 | 460 | . |
| 86 | .. | .. | .. | .. | .. | .. | .. | .. | 307 | 327 | 348 | 368 | 389 | 409 | 430 | 450 | 470 | 491 |
| 88 | .. | .. | .. | .. | .. | .. | .. | .. | 314 | 335 | 356 | 377 | 398 | 419 | 440 | 461 | 481 | 502 |
| 90 | .. | .. | .. | .. | .. | .. | .. | .. | .. | 342 | 364 | 385 | 407 | 428 | 450 | 471 | 492 | 514 |
| 95 | .. | .. | .. | .. | .. | .. | .. | .. | .. | 362 | 384 | 407 | 429 | 452 | 475 | 497 | 520 | 543 |
| 100 | .. | .. | .. | .. | .. | .. | .. | .. | .. | 381 | 404 | 428 | 452 | 476 | 500 | 523 | 547 | 571 |

## Table III.—Nominal Horse-power of Quadruple Compound Engines.

| Diameter of L.P. Cylinders. | Strokes. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 24 ins. | 27 ins. | 30 ins. | 33 ins. | 36 ins. | 39 ins. | 42 ins. | 45 ins. | 48 ins. | 51 ins. | 54 ins. | 57 ins. | 60 ins. | 63 ins. |
| ins. | | | | | | | | | | | | | | |
| 36 | 82 | 92 | 103 | 113 | 123 | 133 | 144 | 154 | .. | .. | .. | .. | .. | .. |
| 38 | 86 | 97 | 108 | 119 | 130 | 141 | 152 | 163 | .. | .. | .. | .. | .. | .. |
| 40 | 91 | 102 | 114 | 125 | 137 | 148 | 160 | 171 | .. | .. | .. | .. | .. | .. |
| 42 | 96 | 108 | 120 | 132 | 144 | 156 | 168 | 180 | 192 | .. | .. | .. | .. | .. |
| 44 | .. | 113 | 125 | 138 | 150 | 163 | 176 | 188 | 201 | .. | .. | .. | .. | .. |
| 46 | .. | 118 | 131 | 144 | 157 | 170 | 184 | 197 | 210 | .. | .. | .. | .. | .. |
| 48 | .. | 123 | 137 | 150 | 164 | 178 | 192 | 205 | 219 | 233 | .. | .. | .. | .. |
| 50 | .. | .. | 142 | 157 | 171 | 186 | 200 | 214 | 228 | 242 | 257 | .. | .. | .. |
| 52 | .. | .. | 148 | 163 | 178 | 193 | 208 | 223 | 237 | 252 | 267 | .. | .. | .. |
| 54 | .. | .. | 154 | 169 | 185 | 200 | 216 | 231 | 247 | 262 | 277 | 293 | 308 | .. |
| 56 | .. | .. | 160 | 176 | 192 | 208 | 224 | 240 | 256 | 272 | 288 | 304 | 320 | .. |
| 58 | .. | .. | .. | 182 | 198 | 215 | 232 | 248 | 265 | 281 | 298 | 315 | 331 | .. |
| 60 | .. | .. | .. | 188 | 205 | 223 | 240 | 257 | 274 | 291 | 308 | 326 | 343 | .. |
| 62 | .. | .. | .. | 194 | 212 | 230 | 248 | 265 | 283 | 301 | 318 | 337 | 354 | .. |
| 64 | .. | .. | .. | 201 | 219 | 238 | 256 | 274 | 293 | 311 | 329 | 348 | 366 | .. |
| 66 | .. | .. | .. | 207 | 226 | 245 | 264 | 283 | 302 | 321 | 339 | 358 | 377 | 396 |
| 68 | .. | .. | .. | 213 | 232 | 252 | 272 | 291 | 311 | 330 | 349 | 369 | 388 | 408 |
| 70 | .. | .. | .. | 220 | 240 | 260 | 280 | 300 | 320 | 340 | 360 | 380 | 400 | 420 |
| 72 | .. | .. | .. | .. | 247 | 267 | 288 | 308 | 329 | 349 | 370 | 391 | 411 | 432 |
| 74 | .. | .. | .. | .. | 254 | 275 | 296 | 317 | 338 | 359 | 380 | 402 | 423 | 444 |
| 76 | .. | .. | .. | .. | 260 | 282 | 304 | 325 | 347 | 369 | 390 | 412 | 434 | 456 |
| 78 | .. | .. | .. | .. | 267 | 290 | 312 | 334 | 356 | 379 | 401 | 423 | 446 | 468 |
| 80 | .. | .. | .. | .. | 274 | 297 | 320 | 342 | 365 | 388 | 411 | 434 | 457 | 480 |
| 82 | .. | .. | .. | .. | 281 | 304 | 328 | 351 | 374 | 398 | 421 | 445 | 468 | 492 |
| 84 | .. | .. | .. | .. | 287 | 311 | 336 | 359 | 383 | 407 | 431 | 456 | 479 | 504 |

## Summary Rules for Estimating Horse-power.

1. Estimated I.H.P. $= \frac{D^2\sqrt{P+15}\times R}{Z} \times \frac{S}{12}$ (Seaton, *v.* "Manual," p. 197).
2. Nominal I.H.P. (Screw Engines) $= \frac{(D^2\sqrt{S}+3H)\ \sqrt[3]{P}}{100}$ (N.E. Coast Inst. E. and S.).
3. Nominal I.H.P. (Paddle Engine) $= \frac{(D^2\sqrt{S+5H})\ \sqrt[3]{P}}{160}$ (N.E. Coast Inst. E. and S.).
4. Estimated I.H.P. $= D^2(S_1+4) \div 262{\cdot}5$ (N.E. Coast Engineers, 1917).

5. Nominal H.P. $(3H + D^2 \sqrt[3]{S}) \times \sqrt[3]{P} \div 700$ (Board of Trade).

6. Nominal H.P. $= \frac{P + C}{K}\left(\frac{D^2\sqrt{S}}{100} + \frac{H}{15}\right)$ (Lloyd's).

7. Nominal H.P. $= D \times S \div X$ (Seaton, *v.* "Manual," p. 196).

8. Nominal H.P. $= D \times S \times \sqrt{P + 15} \div 150$ (Seaton).

$d$ is the diameter of the H.P. cylinder, $d_1$ that of the first intermediate, etc., and D that of the low pressure; S the stroke of piston; all in inches. P the load pressure on safety valves in lbs. per sq. inch; H the heating surface of the boilers in sq. feet; C and K are 340 and 1000 respectively when the boiler pressure is under 160 lbs., and 590 and 1500 when above 160. R the revolutions per minute; and X is 15 for compound engines, 12·5 for triple, and 10·5 for quadruple. Z varies from 85·000 to 105·000 (*v. ante*).

### Table IIIa.—Horse-power in ft.-lbs. of Foreign Countries.

| Country. | British ft.-lbs. per minute. | Ratio of foreign to British. | Kilogrammetres per second. |
|---|---|---|---|
| France . . . . | 32,552 | 0·9865 | 75·000 |
| Prussia . . . | 32,689 | 0·9900 | 75·325 |
| Austria . . . | 33,034 | 1·0010 | 76·119 |
| Saxony . . . | 32,568 | 0·9869 | 75·045 |
| Hanover . . . | 32,705 | 0·9905 | 75·361 |
| German Empire . . | 33,000 | 1·000 | 76·041 |

**Indicated Horse-power.**—The indicated horse-power of an engine may be defined as the measure of work done in the steam cylinder, as deduced from the indicator diagrams, and is equal to (area of piston in square inches × mean pressure in lbs. per square inch × number of feet travelled through by piston, per minute) ÷ 33,000; or,—

**Rule 9.** $$\text{I.H.P.} = \frac{A \times P \times S}{33,000}$$

Piston speed, S, = stroke in feet × 2 × number of revolutions per min.

In the case of engines having more than one steam cylinder, the I.H.P. of each cylinder is determined separately, and the sum of these is the I.H.P. of the engine.

Where accuracy is required, the sectional area of piston rod should be deducted from the areas of piston.

**Shaft Horse-power** is that transmitted by the shaft and is measured by the torque on it. If $T_q$ is the torque in lbs., $\theta$ the angle of torsion on a length of shaft $l$ inches, whose diameter is $d$ inches, then

**Rule 10.** $$\text{S.H.P.} = \frac{\theta^0 \times d^4 \times \text{revs.}}{3{\cdot}27 \times l}.$$

**Nett Horse-power** is that required to overcome the resistance of the ship only, then if R is the resistance in lbs., S the speed in knots per hour, we get

**Rule 11.** $$\text{Nett H.P.} = \frac{\text{R} \times \text{S} \times 6080}{33{,}000 \times 60} = \frac{\text{R} \times \text{S}}{325}.$$

**Tow-rope Horse-power** is that required to tow the ship without a propeller, T is the tension on tow-rope in lbs.

**Rule 12.** $$\text{T}_r\text{H.P.} = \frac{\text{T} \times \text{S}}{325}.$$

**Propeller Horse-power** is that received by the propeller.

**Thrust Horse-power** is that delivered by the propeller for the propulsion of the ship, so that $\text{T}_s$ is the thrust in lbs.

**Rule 13.** $$\text{T}_s\text{H.P.} = \frac{\text{T}_s \times \text{S}}{325}.$$

## Summarising.

(1) I.H.P. is the gross power generated.

(2) S.H.P. is the nett power delivered by the engine to the shafting.

(3) $\text{P}_r$H.P. is that received by the propeller, being S.H.P. *minus* the loss at the thrust and other bearings and in the stern tubes.

(4) $\text{T}_s$H.P. is that delivered to drive the ship, being $\text{P}_r$H.P. less that lost by friction, eddies, etc., in its working.

(5) $\text{T}_r$H.P. is the thrust H.P. less that lost by the augmented resistance set up by the screw, and is practically nett H.P.

To determine the mean pressure from an indicator diagram the following is general practice:—

Let fig. 1 represent a pair of diagrams from the L.P. cylinder of a compound engine. Draw two perpendiculars to the atmospheric line AB, one at each end of and touching the diagrams; divide the space between them into ten equal parts, placing the division marks so that there shall be half a space at each end, and draw a vertical ordinate through each mark; measure off the breadths of the diagrams at each ordinate to the scale corresponding to the indicator spring and figure them on ends of ordinates as shown, keeping the figures referring to each diagram in a separate column.

The sum of each divided by ten gives two mean pressures,—one of which refers to each side of the piston,—and the mean of these two is the mean pressure required.

**Planimeter.**—Where there are many diagrams to be calculated, it is quicker to use a planimeter in place of the method given above. The method is as follows:—Measure the area of the figure by means of the instrument, and divide it by the length AB, when the quotient will be the mean breadth in inches; and this, multiplied by the "scale" of the spring used (the number of pounds required to compress it one inch) will give the mean pressure required.

The Coffin averaging instrument is a form of planimeter specially designed for dealing with indicator diagrams. It leaves on the card two needle pricks, the distance between which is the mean breadth of the figure,—and thus performs mechanically the process of dividing area of figure by length.

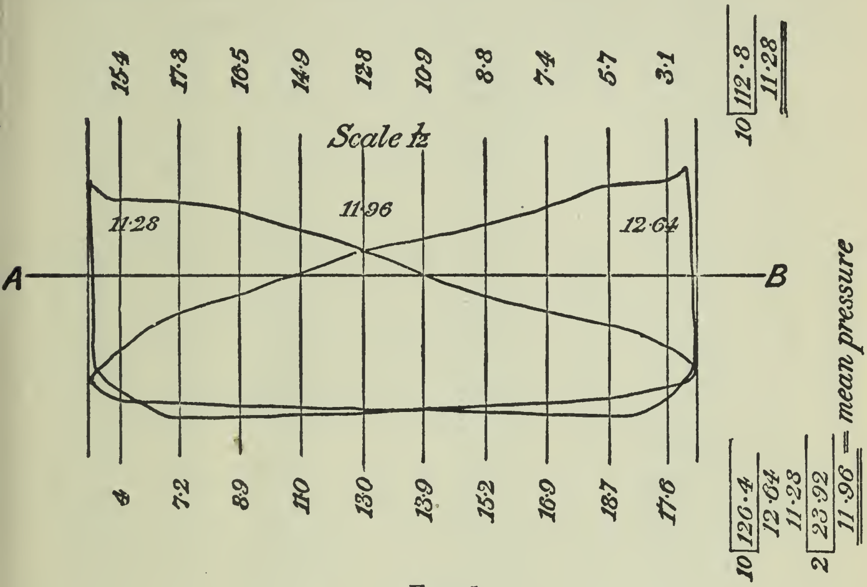

FIG. 1.

The following equivalents may be useful in calculations connected with the above :—

{ Pounds per sq. inch × ·07 = Kilogrammes per sq. centimetre.
{ Kilogrammes per sq. centimetre × 14·22 = Pounds per sq. inch.

{ Foot-pounds × 7·233 = Kilogrammetres.
{ Kilogrammetres × ·138 = Foot-pounds.

{ Horse-power × 1·0139 = Chevaux.
{ Chevaux × ·9863 = Horse-power.

See also "Tables of Pounds per square inch and Kilogrammes per square centimetre," Table CLXXXVIII.

The Continental "Cheval" is equal to 4500 kilogrammetres, or 32,549 foot-pounds per minute, as against 33,000 foot-pounds per minute,—the value of the English horse-power."

The following Table will considerably facilitate the computation of indicated horse-power :—

## Table IV.—Constant Multipliers for I.H.P.

$$\text{Constant} = \frac{\text{area of cylinder}}{33{,}000}\text{ ; and I.H.P} = \text{constant} \times \text{mean press.} \times \text{piston speed.}$$

| Diameter of Cylinder. | Constant. | Diameter of Cylinder. | Constant. | Diameter of Cylinder. | Constant. | Diameter of Cylinder. | Constant. |
|---|---|---|---|---|---|---|---|
| 6 | ·00085 | 16½ | ·00648 | 34 | ·02751 | 60 | ·08569 |
| ¼ | ·00092 | ¾ | ·00668 | ½ | ·02833 | 61 | ·08856 |
| ½ | ·00100 | 17 | ·00688 | 35 | ·02916 | 62 | ·09149 |
| ¾ | ·00108 | ¼ | ·00708 | ½ | ·02999 | 63 | ·09447 |
| 7 | ·00116 | ½ | ·00728 | 36 | ·03084 | 64 | ·09748 |
| ¼ | ·00125 | ¾ | ·00749 | ½ | ·03171 | 65 | ·10055 |
| ½ | ·00134 | 18 | ·00771 | 37 | ·03258 | 66 | ·10368 |
| ¾ | ·00143 | ¼ | ·00792 | ½ | ·03347 | 67 | ·10684 |
| 8 | ·00152 | ½ | ·00814 | 38 | ·03437 | 68 | ·11005 |
| ¼ | ·00162 | ¾ | ·00836 | ½ | ·03528 | 69 | ·11332 |
| ½ | ·00172 | 19 | ·00859 | 39 | ·03620 | 70 | ·11663 |
| ¾ | ·00182 | ¼ | ·00882 | ½ | ·03713 | 71 | ·11998 |
| 9 | ·00192 | ½ | ·00905 | 40 | ·03808 | 72 | ·12339 |
| ¼ | ·00203 | ¾ | ·00928 | ½ | ·03904 | 73 | ·12683 |
| ½ | ·00214 | 20 | ·00952 | 41 | ·04001 | 74 | ·13033 |
| ¾ | ·00226 | ½ | ·01000 | ½ | ·04099 | 75 | ·13388 |
| 10 | ·00238 | 21 | ·01049 | 42 | ·04198 | 76 | ·13748 |
| ¼ | ·00250 | ½ | ·01100 | ½ | ·04299 | 77 | ·14112 |
| ½ | ·00262 | 22 | ·01152 | 43 | ·04401 | 78 | ·14481 |
| ¾ | ·00275 | ½ | ·01205 | ½ | ·04504 | 79 | ·14854 |
| 11 | ·00288 | 23 | ·01259 | 44 | ·04608 | 80 | ·15232 |
| ¼ | ·00301 | ½ | ·01314 | ½ | ·04713 | 81 | ·15615 |
| ½ | ·00314 | 24 | ·01371 | 45 | ·04820 | 82 | ·16003 |
| ¾ | ·00328 | ½ | ·01428 | ½ | ·04927 | 83 | ·16398 |
| 12 | ·00342 | 25 | ·01487 | 46 | ·05036 | 84 | ·16795 |
| ¼ | ·00357 | ½ | ·01547 | ½ | ·05146 | 85 | ·17198 |
| ½ | ·00372 | 26 | ·01609 | 47 | ·05257 | 86 | ·17604 |
| ¾ | ·00387 | ½ | ·01671 | ½ | ·05370 | 87 | ·18016 |
| 13 | ·00402 | 27 | ·01735 | 48 | ·05483 | 88 | ·18432 |
| ¼ | ·00417 | ½ | ·01800 | ½ | ·05598 | 89 | ·18853 |
| ½ | ·00433 | 28 | ·01866 | 49 | ·05714 | 90 | ·19280 |
| ¾ | ·00449 | ½ | ·01933 | ½ | ·05832 | 91 | ·19710 |
| 14 | ·00466 | 29 | ·02001 | 50 | ·05950 | 92 | ·20146 |
| ¼ | ·00483 | ½ | ·02071 | 51 | ·06191 | 93 | ·20587 |
| ½ | ·00500 | 30 | ·02142 | 52 | ·06436 | 94 | ·21030 |
| ¾ | ·00517 | ½ | ·02214 | 53 | ·06685 | 95 | ·21480 |
| 15 | ·00535 | 31 | ·02287 | 54 | ·06940 | 96 | ·21937 |
| ¼ | ·00553 | ½ | ·02362 | 55 | ·07200 | 97 | ·22394 |
| ½ | ·00571 | 32 | ·02437 | 56 | ·07464 | 98 | ·22859 |
| ¾ | ·00590 | ½ | ·02513 | 57 | ·07733 | 99 | ·23328 |
| 16 | ·00609 | 33 | ·02592 | 58 | ·08006 | 100 | ·23799 |
| ¼ | ·00628 | ½ | ·02671 | 59 | ·08285 | 101 | ·24280 |

When the effective pressure on the piston at each point in the stroke is required,—as for instance, to calculate the twisting moment on the crankshaft, — diagrams should be constructed from the indicator diagrams, as follows :—

First, draw the line of no pressure, CD. Then, dealing with one stroke at a time, the curve EFG represents the varying pressures on one side of the piston, whilst the opposing pressures are represented by the curve JKB, which forms a part of the diagram from the other side of the piston.

Draw any ordinate FH, and set off HL = HF − HK ; then L is a point in the required resultant diagram in which any number of other points may be found in a similar way ; at the point H, in the stroke CD, the effective pressure is HL. When the quantity corresponding to HF − HK is minus, it must be set off below the line of no pressure.

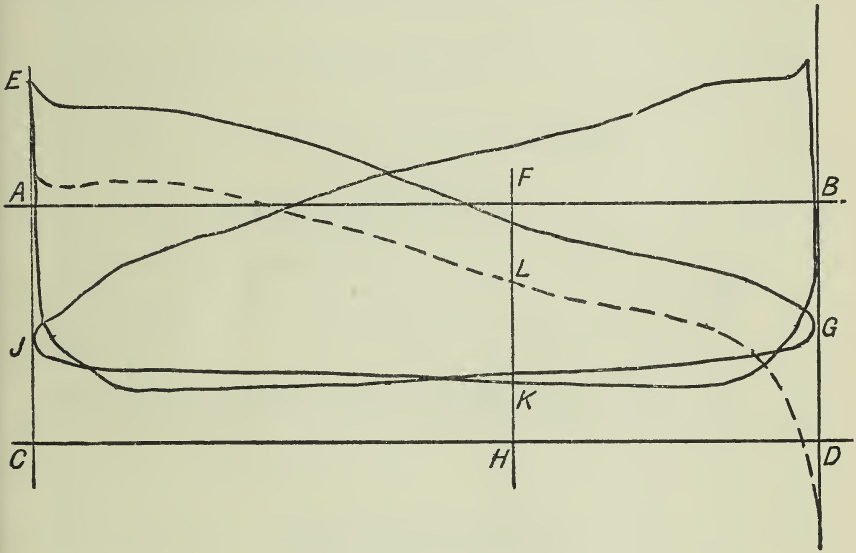

FIG. 2.

The pipes leading from the ends of the cylinder to the indicator should be large, short, of equal length, and as free from bends as possible ; otherwise, there will be loss of area in the diagram, and the apparent I.H.P. less than that really developed.

These pipes vary, in common practice, from $\frac{3}{4}$-inch to 1-inch diameter, according to their length and the piston speed, but are still, beyond doubt, the cause of very perceptible loss of area.

To make accurate tests of engines, to determine water consumption per I.H.P., &c., a separate indicator should be fitted direct to each end of each cylinder. This practice should always be adopted where anything like accuracy is required.

N. B.—The indicator shows only *differences* between the pressures of the steam and of the atmosphere, and not absolute pressures.

**To ascertain the weight of steam** accounted for by any diagram, take a point A in the expansion curve of the diagram (fig. 3), just before release, and measure the absolute pressure there; then take another point B in the compression curve, and measure the absolute pressure here also; from Table CLXXIX. ascertain the weight of a cubic foot of steam at each of the pressures AZ and BZ.

Now calculate the volume, in cubic feet, swept by the piston while travelling through the distance X, and multiply it by the weight per cubic foot at pressure AZ; also calculate the volume corresponding to the travel Y, and multiply it by the weight per cubic foot at pressure BZ; subtract the second product from the first, the remainder will be the number of pounds of steam accounted for by the diagram during the stroke.

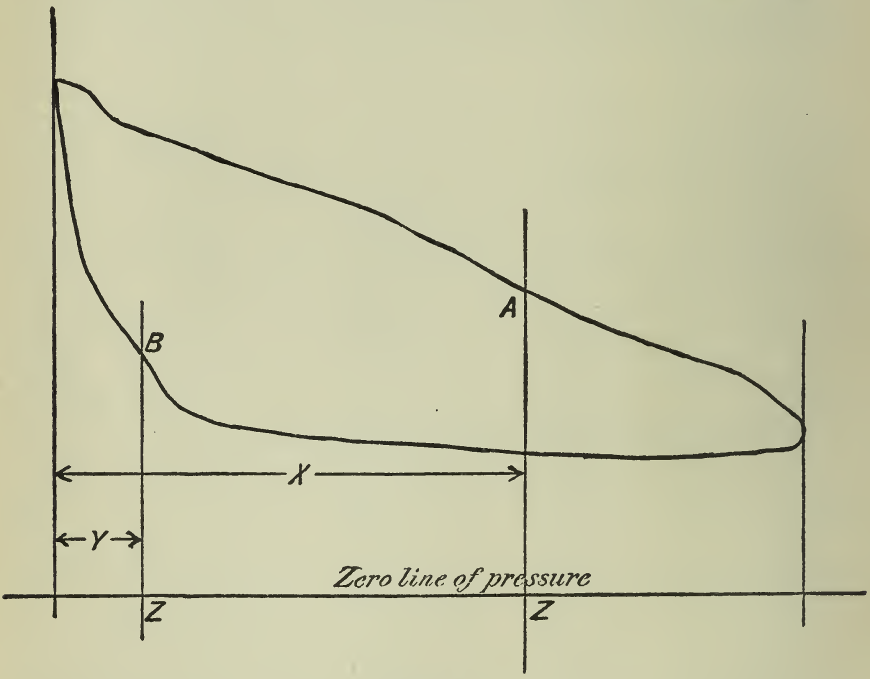

FIG. 3.

A similar calculation from the other diagram of the pair will give the amount of steam accounted for during the return stroke, and the sum of the two—multiplied by the number of revolutions—the amount per minute or per hour.

The clearances need not be considered in these calculations if the points A and B be taken at the same distance above the zero line of pressure.

This method is mainly useful in determining condensation and re-evaporation that occur during passage of steam through a series of cylinders. No conclusion as to economy can be derived from diagrams only. They tell nothing about water that may be present in cylinders. Feed water must be measured to determine real economy of engine.

**Shaft Horse-power** is that transmitted by a shaft, and can be calculated from the *torque* as follows :—

T is the twisting moment or *torque* in inch-pounds.

R, the revolutions made per minute by the shaft.

The work performed per revolution $= \frac{2\pi \times T}{12}$ or 0·5236 T.

**Rule 14.** $$\text{S.H.P.} = \frac{0{\cdot}5236\ T \times R}{33{,}000} = \frac{T \times R}{63{,}000}.$$

The torque on any shaft can be calculated from the angle of twist or torsion by means of the following formulæ :—

$\alpha$ is the arc at a radius $r$ of the angle of torsion $\theta$, $\frac{\alpha}{r} = \beta$ ; $l$ is the length and $d$ the diameter of shaft under observation.

$$\beta = \frac{10{\cdot}2 \times T \times l}{M_r \times d^4} \quad \text{(Rankine).}$$

$$\frac{\theta}{360} = \frac{\alpha}{2\pi r} \text{ or } = \frac{\beta}{2\pi}.$$

That is $$\beta = \frac{\theta \times 2\pi}{360} = \frac{\theta}{57{\cdot}3}.$$

Then $$\frac{\theta}{57{\cdot}3} = \frac{10{\cdot}2 \times T \times l}{M_r \times d^4},$$

or (i) $$\theta = \frac{584 \times T \times l}{M_r \times d^4} \text{ for solid shafts,}$$

(ii) $$\theta = \frac{584\ T \times l}{M_r(d^4 - d_1{}^4)}$$ for hollow shafts when $d_1$ is the diameter of the bore.

That is $$T = \frac{\theta \times d^4 \times M_r}{584 \times l}.$$

$M_r$ is the modulus of stiffness or rigidity of the material, which for steel generally is 10 to 12 millions. With steel shafts of best make, experiments have shown the value of $M_r$ to be 11,750,000 for solid, and

12,150,000 for hollow. In everyday practice, 11,250,000 is taken for solid steel shafts:—

then $$T = \frac{\theta \times d^4 \times 11,250,000}{584\ l} = 19,264 \frac{\theta \times d^4}{l}.$$

Substituting this value of T in the formula for S.H.P.

**Rule 15.** $$\text{S.H.P.} = 19,264\ \frac{\theta \times d^4}{l} \times \frac{R}{63,000} = \frac{\theta \times d^4 \times R}{3{\cdot}27 \times l} \text{ solid shafts.}$$

**Rule 15a.** $$\text{S.H.P.} = \frac{\theta(d^4 - d_1{}^4)R}{3{\cdot}27 \times l} \text{ for hollow shafts.}$$

**Torsion Meters** are the instruments employed to indicate the angular displacement of a definite length of shaft (usually 40 inches), when the shaft is receiving and transmitting power. They are all designed on the same fundamental principle, and consist of two cast-iron sleeves made in halves so as to go on the shaft, and be keyed to it a definite distance apart; one is short and the other long; each has a disc of considerable diameter at the ends adjoining, so that as the shaft twists the discs' edges are displaced with respect to one another by the amount $\alpha$, which may be used instead of $\theta°$, if desired, for calculating S.H.P. thus:—

**Rule 15b.** $$\text{S.H.P.} = \frac{17{\cdot}5 \times d^4 \times R \times \alpha}{r \times l}.$$

There are many forms of torsion meter which may be classed by the methods adopted for indicating $\theta$ or $\beta$:—

(i) Those having differential levers whereby the distortion is magnified, such as Professor Föttinger's.
(ii) Those which accomplish the same end by wheel and pinion gearing, as in the Denny-Edgecombe machine.
(iii) Those without sleeves, but with counter shafts coupled at their extreme ends to the shaft by driving gear and to one another by screw and nut, which cause longitudinal displacement, which is magnified by a lever; as in the Collie meter.
(iv) Those which magnify the arc of the angle of torque by means of a beam of light reflected from a mirror, which is twisted through angle $\theta$; as in the Hopkinson-Thring instrument, which entails the use of the least amount of gearing.
(v) Others, as the Bevis-Gibson, where a beam of light is made to pass through slits in discs at a considerable distance apart; which becomes blinded on angular distortion taking place, and visible again after an angular movement of the sighting hole corresponding to $\theta$.

The registering is made mechanically and automatically by the *Föttinger* on a sheet of paper secured on a fixed drum surrounding the shaft, so that the magnitude of the angle or arc can be measured at any point by the distance from the base line made when the shaft is transmitting no power.

The *Denny-Edgecombe* machine shows the distortion on a dial which, however, is in motion on the shaft, but by a most ingenious electrical arrangement its indications are given on another dial fixed in any convenient part of the ship. It can be and often is made to automatically register its indications on a diagram, which show us the vagaries in torque of a reciprocator, or the mean torque of it or of a turbine.

The *Hopkinson-Thring* mirrors throw a point or line of light on a fixed scale, from which at any time the angle may be read off; it also registers its own zero line continuously, and by an arrangement can indicate the variations in torque of a reciprocator.

The *Bevis-Gibson* measures the torque by the amount of displacement required in the registering apparatus to bring the light holes in line.

A simple formula for each ship can be made and used for S.H.P. by simply multiplying it by $\theta$ and revolutions; in fact, a table may be made by including the revolutions, so that from it a co-efficient may be taken and multiplied by $\theta$ only. Thus if $l$ is 40 and the diameter of the shaft 12 ins., then

**Rule 16**. $\text{S.H.P.} = 158{\cdot}5\,\theta \times \text{R}.$

And if at 400 revolutions $\theta$ is $0{\cdot}165°$, S.H.P. = 10,461.

## EFFICIENCY OF MARINE MACHINERY.

*Efficiency* is expressed by the fraction of the whole work or energy generated or supplied that is usefully employed for the purpose for which it was designed. Hence,

*Efficiency of the boiler* is the available heat transmitted to the engine as a fraction of that given out by the fuel.

*Efficiency of the steam* is the fraction of the total heat of evaporation that is capable of being converted into work.

*The thermal efficiency* of an engine is expressed by the fraction of the work capable of being done by the steam or gas that actually is done by it.

*The mechanical efficiency* is the fraction of the work developed in the generator that is actually passed on from it for external consumption.

*The general efficiency* of a steam or gas engine is the fraction that the output of energy is of that available from the fuel.

*Propeller efficiency* is the fraction of the power delivered to it that is devoted to thrust power.

*Propulsive efficiency* is the relation of the nett or tow-rope H.P. to the gross H.P. developed in the propelling engines; strictly speaking, it should be taken to the H.P. delivered to the propeller shafting, which can now be ascertained by torsion meters.

*Boiler efficiency* is measured by taking the number of pounds of steam actually evaporated as the numerator and the thermal value of the fuel divided by the total heat of evaporation under the same conditions as denominator. For example, the average thermal value of the ordinary good Welsh steam coal is 15,500 B.T.U. The total heat of evaporation from 110° and at 341° (120 lbs. pressure abs.) is 1108 B.T.U., then

$$\text{greatest possible evaporation} = \frac{15{,}500}{1108} = 14 \text{ lbs.}$$

A boiler using this fuel evaporates 10·3 lbs. of steam per lb. of fuel.

$$\text{The efficiency} = \frac{10{\cdot}3}{14} \text{ or } 0{\cdot}736.$$

*Efficiency of steam* by Rankine's formula is as follows:—

$$E = \frac{\text{effective mean pressure in cylinders}}{\text{mean pressure} + \text{rate of expansion} \times pr^{-17}/_{16}}.$$

*Thermal efficiency* is expressed, then, where $T_1$ is the absolute temperature at entry and $T_2$ that at exit or exhaust.

**Rule 17.** Then thermal efficiency $= \dfrac{T_1 - T_2}{T_1}$.

For example, an engine works with steam at 180 lbs. pressure abs. at the H.P. valve and rejects into a condenser with a vacuum of 28 ins. Here $T_1 = 373° + 461° = 834°$.
$T_2 = 90° + 461° = 551°$.

$$\text{Thermal efficiency} = \frac{834 - 551}{834} = 0{\cdot}315.$$

### The Maximum Thermal Efficiency of any Condensing Steam Engine with a Temperature of 60° F. in the Condenser.

| Working press. abs. | 40 | 80 | 100 | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 260 | 280 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Efficiency . . | 0·284 | 0·326 | 0·339 | 0·350 | 0·360 | 0·368 | 0·375 | 0·381 | 0·387 | 0·393 | 0·398 | 0·404 |

*The mechanical efficiency* of engines generally is ascertained by comparing the output H.P. as measured by a brake or other mechanical means. This is not possible with engines of great power. Since the introduction of the turbine as a marine motor, the torsion meter has been invented and used to indicate S.H.P. in reciprocators. Before this experiments had been made to ascertain the friction H.P. by running engines without their propellers at various rates of revolution and deducting it from the I.H.P. at the corresponding revolutions coupled up. The following has been ascertained by one or other of these methods:—

**Rule 18.** Mechanical efficiency $= \dfrac{\text{S.H.P. or B.H.P.}}{\text{I.H.P.}}$.

Large quadruple engines by Denny Bros., efficiency $\dfrac{\text{S.H.P.}}{\text{I.H.P.}} = 92$ to 94 per cent.

Triple compound engines 900 I.H.P. by Central Marine Eng. Co., $\dfrac{\text{I.H.P.} - \text{F.H.P.}}{\text{I.H.P.}} = 95$ per cent.

Several large engines (triples) made in Germany showed an efficiency from 88·5 per cent. to 93·5 in the largest (4500 I.H.P.).

Triple compound engines, 300 I.H.P., by Earles S. & E. Co. showed 84·9 per cent., and similar engines 480 I.H.P. 90·2.

Compound engines of a torpedo boat by Mr A. F. Yarrow, 255 I.H.P., showed by dynamometric trials 92·3 per cent., and as much as 76·6 per cent. when developing only 38·5 I.H.P.

Triple compound engines with *forced lubrication* by Belliss & Morcom gave by electric output and brake trials 93·15 per cent. at 429 I.H.P., and as much as 98·8 per cent. at 218 I H.P.

Marine engine losses vary very nearly with the revolutions but the power as the cube, the efficiency at low speeds is then less than at high ones with the same engine; further, the efficiency of small engines, running on similar conditions as to pressure, rates of expansion, etc., as large ones, is lower. On the other hand, the smaller the engine is for the power to be developed the higher will be its efficiency mechanically.

*General efficiency of steam engines* is gauged by the water or steam consumption per I.H.P. The following schedule gives the greatest possible output of work by a pound of steam under various conditions, and the corresponding horse-power:—

If a steam engine consumes $x$ lbs. of steam per H.P. hour, or $\dfrac{x}{60}$ per H.P. minute, then $x = 60$ I.H.P., or, 1 lb. produces $\dfrac{60}{x}$ I.H.P.

If X is the greatest possible output under similar conditions.

**Rule 19.** General efficiency of engines $= \dfrac{60}{x} \div \text{X}$ or $\dfrac{60}{\text{X}x}$.

*General efficiency of all engines* must, however, be commercially dealt with differently, as fuel is the serious factor in any installation and must form the basis of comparison of a steam with an internal combustion engine. In this case the thermal value of a H.P. is 33,000 ÷ 778 or 42·4 B.T.U. The value per hour is 2544 B.T.U. The Welsh coal, with its calorific value of 15,500 B.T.U., is equal to $\dfrac{15{,}500}{2544}$ or 6·10 H.P.

If an engine uses 1·3 lbs. per I.H.P. hour,

its efficiency will be $\dfrac{1}{6{\cdot}1 \times 1{\cdot}3} = 0{\cdot}126$ or only 12·6 per cent.

The oil engine using 0·45 lb. of fuel, whose calorific value is 18,800 B.T.U., will have an efficiency as ascertained in the same way—

$$1 \div \frac{18,000}{2544} \times 0{\cdot}45 = 0{\cdot}30 \text{ or } 30 \text{ per cent.}$$

**Rule 20.** General efficiency of machinery installations
$= 2544 \div$ B.T.U. value of fuel × weight consumed per H.P. hour.

From observations made at the tests of electric generating engines it is found that at constant revolutions the power required to overcome the friction of the engine itself is the same whatever the load may be; and with varying revolutions and varying loads the friction *per revolution* is nearly constant; it is therefore probable that with marine engines total friction varies nearly directly as the revolutions.

Now, as the total I.H.P. varies as the cube of the revolutions, it will be seen that the mechanical efficiency of a particular marine engine is at a maximum at full speed.

**Froude's Method.**—As there is considerable difficulty in experimentally determining the power absorbed in overcoming the friction of a marine engine, the following graphic method is of interest:—

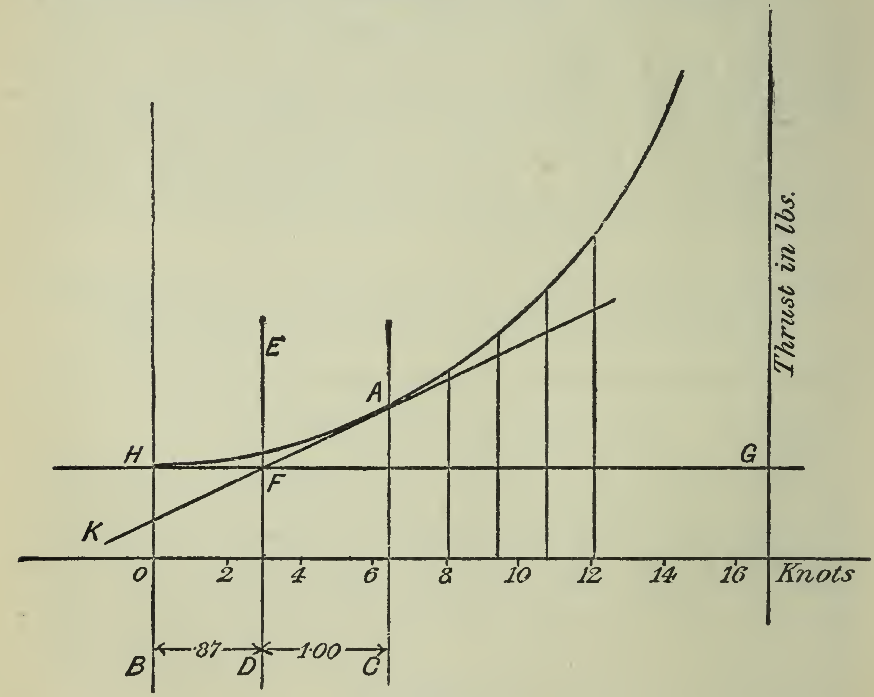

FIG. 4.

A series of progressive trials having been carried out, and the results carefully recorded—calculate the indicated thrusts (from the formula—Indicated thrust $=\frac{\text{I H.P.} \times 33,000}{\text{P} \times \text{R}}$, where P is pitch of propeller in feet, and R revolutions per minute) for each speed, and set them up as ordinates from a base line on which the speeds are set off—as shown in fig. 4. Then, supposing A to be the lowest known point on the curve, draw the tangent KA ; divide BC at D so that BC = 1·87 DC, and through D draw the vertical line DE, cutting the tangent in F ; and through F draw HG parallel to the base line. The height OH then represents the constant friction of the engine, and the point H is the vertex of the thrust curve which may now be completed.

The lengths of the ordinates intercepted between HG and the thrust curve represent power expended in overcoming ship's nett resistance, augment of resistance due to propeller, and friction of propeller blades, and are proportional to the ship's resistance,

It is also very difficult to determine the frictional resistance of any pair of parts of an engine, owing to the difficulty of reproducing exact conditions of actual work in an experimental apparatus, so here again values can only be estimated from those obtained by experiments made under conditions approximating to those of an actual engine.

The experiments of Tower (made for the Institution of Mechanical Engineers, 1883–91) and of Dewrance (1896) are instructive, as the results obtained were most important ; but their tests were all made on journals under constant load, and therefore under very different conditions to those prevailing in the crank bearings of reciprocating engines, and experience has shown the latter to be capable of carrying much greater pressures than bearings subject to constant load.

Tower and Dewrance's experiments may be summarised as follows :—

1. When a bearing is running satisfactorily the surfaces are not in metallic contact, but are separated by a film of oil.

2. Actual measurement by pressure gauges showed that near centre of brass oil was under a pressure somewhat greater than mean pressure per square inch (*i.e.* total load ÷ dia. × length of journal) carried by bearing at the time, and that pressure decreased as ends and sides of bearing surface were approached, and became zero at edges.

3. The drilling of an oil hole at centre of crown of brass allowed oil to be forced out, caused pressure to fall, and diminished load that bearing could carry. These results made it clear that oil should be introduced to bearings of this type where pressure is least, and that no openings should be made near centre of bearing surface, where pressure is greatest. It was found necessary to lubricate experimental journal by means of an oil bath, because, when oil holes in top of brass were used, results were very irregular and frictional resistance was about four times that registered when oil bath was in use.

4. So long as lubrication remained efficient the frictional resistance under varying loads remained constant, *i e.* was unaffected by pressure per square inch to which bearing was loaded.

5. The frictional resistance varied very nearly as the square root of the rubbing velocity.

6. The temperature at which the bearing was run was found to be most important, one experiment, made with lard oil under a pressure of 100 lbs. per square inch, showing that the frictional resistance was three times greater at 60° than at 120°. Apparently there is a most suitable temperature for each lubricant at each pressure.

For a temperature of 90° the above results are very accurately expressed by the formula—

$$F = 20C \times \frac{\sqrt{V}}{P},$$

where F = friction factor or co-efficient, V = rubbing velocity in ft. per min., P = nominal pressure in lbs. per sq. in., C = ·0014 for sperm oil, ·0015 for rape, ·0018 for mineral, and ·0019 for olive oil. P had a range of about 100 to 500 lbs. per square inch.

Frictional resistance depends on—(*a*) Velocity of rubbing; (*b*) intensity of pressure; (*c*) temperature; (*d*) lubricant; (*e*) nature of rubbing, *i.e.* continuous or constantly reversed; (*f*) form of surface, *i.e.* curved, like a bearing, or flat; (*g*) material and condition of surface; (*h*) extent of contact, *i.e.* all over a surface, or only on a line; (*j*) method of application of lubricant.

As regards (*g*), Dewrance's experiments appear to show that the composition of the alloys used had, other things being equal, little or no effect on results obtained,—a result difficult to reconcile with common experience to the contrary with ordinary engine bearings.

As regards (*h*), reduction in surface appears to reduce frictional resistance (as in the case of a railway carriage wheel when "skidded"), so that surfaces should not greatly exceed those found to give reasonable length of life under usual conditions.

Dewrance found (*j*) most important, and by careful arrangements succeeded in carrying loads of well over one ton per square inch, whereas with defective (but very usual) arrangements the same bearing would not carry one-tenth of that load. He found it necessary to ease away the brasses at the sides in a very gradual manner and to admit the oil at the sides, or points of least pressure, whence it was carried forward by the revolving journal into the gradually narrowing clearance spaces and towards the points of greatest pressure.

Any sudden contraction of this clearance space, or any ridge tending to scrape the oil from the surface of the journal, was found to lower seriously the load-carrying power of the bearing.

The fact that a crosshead shoe will not usually work satisfactorily with a load of more than about 50 lbs. per square inch, whilst a bearing can easily be made to carry ten times as much, still waits an explanation. Difficulty of lubrication, and the continual reversal of the motion, will no doubt partly explain the anomaly; but, even if all the rubbing surfaces about an engine were of equal smoothness, worked at same temperature, with same lubricant, and at same speed of rubbing, it would still be almost impossible to fix the various values of the friction factor. And as to frictional losses caused by tightness of piston rings, and of gland packings, and by screwing up of bearings,—they may be anything, and cannot even be estimated.

The efficiency of the same engine may therefore vary greatly at different times, and the frictional resistance of a bearing (say) may be almost anything between that due to the friction of solid on solid and that due to liquid friction. The laws governing the friction of solids are as follows:—

(*a*) The frictional resistance varies directly as the load.

(*b*) It is independent of the extent of surface in contact.

(*c*) It tends to diminish with an increase of velocity above a certain limit.

## THE THERMAL AND POWER VALUE OF A POUND OF STEAM.

### Maximum Output from 1 lb. of Steam in B.T.U. and Horse-power.

The following Tables—V., VI., and VII.—give the amount of work that is theoretically possible under the different conditions stated. Table VII. is a general one giving the output when steam is expanded from an actual pressure $p_1$ to a pressure $p_2$, at which it discharges; this is the case of a turbine during its various stages and throughout. Table V. gives the output when steam is expanded from a pressure $p_1$, expands to a pressure $p_2$, and then exhausts to a condenser in which the pressure is one pound—that is, the vacuum is 28 inches, which is the case of the ordinary reciprocating condensing engine. Table VI. gives the output when exhausting to the atmosphere, as in the case of locomotives and other engines which have no condenser. It will be seen that a turbine supplied with steam at 100 lbs. pressure (absolute) can do the same work as a triple compound reciprocator expanding steam of 150 lbs. to 5 lbs., or one of 200 lbs. expanding to 7 lbs. And a locomotive would require 55 per cent. more steam at 200 lbs. pressure to do the same work, supposing the efficiency of the means is the same in each case. It will also be seen that steam at 100 lbs. has a 19 per cent. higher potential than that of 50 lbs; that 150 is 8·44 per cent. higher than 100; that 200 is 5·77 per cent. higher than 150; and 250 is 4·82 per cent. higher than 200. Also that the 250 lbs. of the new quadruple has a potential 7·41 per cent. greater than the 175 lbs. of the older triples.

Table V.—Maximum Work possible from One Pound of Steam of Pressure $p_1$, Expanding Adiabatically to Pressure $p_2$, and Exhausting to Condenser at 1 lb. Pressure.

| Initial pressure. | 100 lbs. | | 125 lbs. | | 150 lbs. | | 175 lbs. | | 200 lbs. | | 225 lbs. | | 250 lbs. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Termina $p_2$ | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. |
| 1·0 lb. | 288·0 | 6·80 | 303·0 | 7·14 | 314·5 | 7·42 | 324·0 | 7·64 | 332·5 | 7·84 | 340·6 | 8·03 | 348·0 | 8·21 |
| 2·0 lbs. | 286·8 | 6·76 | 301·8 | 7·09 | 313·3 | 7·26 | 322·4 | 7·61 | 331·3 | 7·81 | 339·3 | 8·00 | 346·8 | 8·18 |
| 3·0 ,, | 277·2 | 6·54 | 292·6 | 6·90 | 309·2 | 7·17 | 313·5 | 7·40 | 322·2 | 7·60 | 330·2 | 7·80 | 337·2 | 7·95 |
| 4·0 ,, | 266·2 | 6·28 | 281·9 | 6·65 | 294·0 | 6·93 | 303·2 | 7·15 | 312·0 | 7·36 | 320·2 | 7·55 | 328·2 | 7·74 |
| 5·0 ,, | 258·0 | 6·09 | 273·5 | 6·45 | 285·0 | 6·72 | 295·0 | 6·96 | 304·0 | 7·17 | 312·0 | 7·36 | 320·0 | 7·55 |
| 6·0 ,, | 249·9 | 5·90 | 266·9 | 6·27 | 278·0 | 6·55 | 287·9 | 6·80 | 297·4 | 7·01 | 306·8 | 7·23 | 313·9 | 7·40 |
| 7·0 ,, | 243·9 | 5·75 | 259·3 | 6·11 | 271·9 | 6·42 | 281·8 | 6·64 | 290·8 | 6·86 | 300·3 | 7·09 | 307·8 | 7·26 |
| 8·0 ,, | 237·5 | 5·61 | 253·5 | 5·96 | 265·5 | 6·26 | 276·8 | 6·53 | 285·5 | 6·73 | 294·5 | 6·94 | 302·5 | 7·13 |
| 9·0 ,, | 232·0 | 5·47 | 248·0 | 5·85 | 260·5 | 6·14 | 271·3 | 6·40 | 280·7 | 6·62 | 289·1 | 6·82 | 297·0 | 7·00 |
| 10 ,, | 226·9 | 5·35 | 242·1 | 5·71 | 254·0 | 6·00 | 265·5 | 6·26 | 275·9 | 6·51 | 284·1 | 6·70 | 292·1 | 6·89 |
| 11 ,, | 222·1 | 5·24 | 237·1 | 5·59 | 249·8 | 5·89 | 260·7 | 6·15 | 270·5 | 6·38 | 279·1 | 6·58 | 287·1 | 6·77 |
| 12 ,, | 217·0 | 5·12 | 233·0 | 5·49 | 245·5 | 5·80 | 256·1 | 6·04 | 265·4 | 6·26 | 275·0 | 6·48 | 283·0 | 6·67 |
| 13 ,, | 212·2 | 5·00 | 228·6 | 5·39 | 240·7 | 5·69 | 251·6 | 5·93 | 261·1 | 6·16 | 270·7 | 6·39 | 278·7 | 6·57 |
| 14 ,, | 207·4 | 4·90 | 224·1 | 5·28 | 236·4 | 5·60 | 247·2 | 5·83 | 257·3 | 6·07 | 266·4 | 6·29 | 274·4 | 6·47 |
| 15 ,, | 204·0 | 4·81 | 220·0 | 5·19 | 232·0 | 5·47 | 243·5 | 5·74 | 253·0 | 5·97 | 262·0 | 6·18 | 270·0 | 6·37 |
| 16 ,, | 200·6 | 4·73 | 216·6 | 5·11 | 228·6 | 5·39 | 239·6 | 5·64 | 249·9 | 5·89 | 258·8 | 6·10 | 266·4 | 6·28 |
| 17 ,, | 197·4 | 4·65 | 213·3 | 5·03 | 225·0 | 5·30 | 236·0 | 5·56 | 246·5 | 5·81 | 255·3 | 6·02 | 262·6 | 6·19 |
| 18 ,, | 194·3 | 4·58 | 210·1 | 4·95 | 220·0 | 5·23 | 232·9 | 5·49 | 243·3 | 5·74 | 252·0 | 5·94 | 259·9 | 6·11 |
| 19 ,, | 191·3 | 4·50 | 207·2 | 4·88 | 219·4 | 5·17 | 230·0 | 5·42 | 240·5 | 5·67 | 249·3 | 5·88 | 257·3 | 6·06 |
| 20 ,, | 188·4 | 4·44 | 204·4 | 4·82 | 217·3 | 5·12 | 228·0 | 5·38 | 238·0 | 5·61 | 247·0 | 5·82 | 255·4 | 6·02 |
| 25 ,, | 176·0 | 4·15 | 191·3 | 4·51 | 205·0 | 4·83 | 215·9 | 5·09 | 225·4 | 5·31 | 234·2 | 5·52 | 243·0 | 5·73 |
| 30 ,, | 164·8 | 3·89 | 180·0 | 4·24 | 194·5 | 4·59 | 205·0 | 4·83 | 214·9 | 5·07 | 223·4 | 5·27 | 232·0 | 5·47 |
| 35 ,, | 155·2 | 3·66 | 169·6 | 4·00 | 184·2 | 4·34 | 195·2 | 4·60 | 205·2 | 4·84 | 214·2 | 5·05 | 222·2 | 5·24 |
| 40 ,, | 146·2 | 3·45 | 160·5 | 3·80 | 175·1 | 4·13 | 186·0 | 4·40 | 196·0 | 4·62 | 205·0 | 4·81 | 213·0 | 5·02 |

Table VI.—Maximum Work possible from One Pound of Steam of Pressure $p_1$, Expanding Adiabatically to Pressure $p_2$, and Exhausting against 15 lbs.

| Initial pressure. | $p_1$ 100 lbs. | | $p_1$ 125 lbs. | | $p_1$ 150 lbs. | | $p_1$ 175 lbs. | | $p_1$ 200 lbs. | | $p_1$ 225 lbs. | | $p_1$ 250 lbs. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Terminal $p_2$. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | BT .U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. |
| 15 lbs. | 137·0 | 3·23 | 153·1 | 3·61 | 165·0 | 3·89 | 176·5 | 4·16 | 186·5 | 4·40 | 195·4 | 4·61 | 203·3 | 4·80 |
| 20 ,, | 135·7 | 3·20 | 152·3 | 3·60 | 164·3 | 3·88 | 176·0 | 4·15 | 186·1 | 4·39 | 195·1 | 4·60 | 203·1 | 4·79 |
| 25 ,, | 132·6 | 3·13 | 149·6 | 3·53 | 162·6 | 3·83 | 173·6 | 4·09 | 183·6 | 4·33 | 192·6 | 4·54 | 201·6 | 4·75 |
| 30 ,, | 128·7 | 3·04 | 145·4 | 3·43 | 158·4 | 3·74 | 169·4 | 4·00 | 179·4 | 4·23 | 189·4 | 4·47 | 197·5 | 4·66 |
| 35 ,, | 123·8 | 2·92 | 140·6 | 3·32 | 153·8 | 3·63 | 165·1 | 3·90 | 175·1 | 4·13 | 184·4 | 4·35 | 192·1 | 4·53 |
| 40 ,, | 119·6 | 2·82 | 135·6 | 3·20 | 148·6 | 3·50 | 160·6 | 3·79 | 170·6 | 4·02 | 179·6 | 4·23 | 187·6 | 4·42 |
| 45 ,, | 114·5 | 2·70 | 130·0 | 3·07 | 143·5 | 3·40 | 155·0 | 3·66 | 165·0 | 3·89 | 174·0 | 4·10 | 182·0 | 4·30 |
| 50 ,, | 108·0 | 2·55 | 124·4 | 2·93 | 138·8 | 3·27 | 150·6 | 3·54 | 160·5 | 3·78 | 169·0 | 3·99 | 177·0 | 4·18 |
| 55 ,, | 103·2 | 2·43 | 119·7 | 2·82 | 133·9 | 3·16 | 145·4 | 3·43 | 155·4 | 3·67 | 164·4 | 3·88 | 172·4 | 4·07 |
| 60 ,, | 98·2 | 2·32 | 114·7 | 2·71 | 130·4 | 3·08 | 141·9 | 3·35 | 151·4 | 3·57 | 160·4 | 3·78 | 168·4 | 3·97 |
| 65 ,, | 93·4 | 2·20 | 110·2 | 2·61 | 127·2 | 3·00 | 138·2 | 3·26 | 147·7 | 3·48 | 155·6 | 3·67 | 164·2 | 3·88 |
| 70 ,, | 88·7 | 2·09 | 106·8 | 2·52 | 123·8 | 2·92 | 134·8 | 3·18 | 144·1 | 3·40 | 152·1 | 3·59 | 160·1 | 3·78 |
| 75 ,, | 84·2 | 1·99 | 104·6 | 2·46 | 120·0 | 2·83 | 130·4 | 3·08 | 140·3 | 3·31 | 148·2 | 3·50 | 156·2 | 3·69 |
| 80 ,, | 79·9 | 1·89 | 101·2 | 2·38 | 114·8 | 2·71 | 126·5 | 2·99 | 136·6 | 3·22 | 144·9 | 3·42 | 151·7 | 3·59 |
| 85 ,, | 76·3 | 1·80 | 98·1 | 2·31 | 110·9 | 2·61 | 122·9 | 2·90 | 133·3 | 3·14 | 141·7 | 3·34 | 148·7 | 3·50 |
| 90 ,, | 72·9 | 1·72 | 94·8 | 2·24 | 108·4 | 2·56 | 119·9 | 2·83 | 129·5 | 3·06 | 138·3 | 3·26 | 145·3 | 3·42 |
| 95 ,, | 70·2 | 1·65 | 91·4 | 2·16 | 104·4 | 2·46 | 116·9 | 2·76 | 126·5 | 2·98 | 135·3 | 3·19 | 141·7 | 3·34 |
| 100 ,, | 68·3 | 1·61 | 88·3 | 2·09 | 102·3 | 2·40 | 113·3 | 2·67 | 123·3 | 2·90 | 131·9 | 3·11 | 138·4 | 3·27 |

## Table VII.—Maximum Work possible from One Pound of Steam of Pressure $p_1$, Expanding Adiabatically to Pressure $p_2$, and Exhausting against Pressure $p_2$.

| $p_1$. | 25 lbs. | | 50 lbs. | | 75 lbs. | | 100 lbs. | | 125 lbs. | | 150 lbs. | | 175 lbs. | | 200 lbs. | | 225 lbs. | | 250 lbs. | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $p_2$. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. | B.T.U. | H.P. |
| 1·0 | 188 | 4·43 | 243 | 5·73 | 269 | 6·34 | 289 | 6·81 | 303 | 7·14 | 314 | 7·41 | 324 | 7·64 | 332 | 7·83 | 340 | 8·02 | 348 | 8·21 |
| 2·0 | 163 | 3·85 | 209 | 4·93 | 234 | 5·52 | 254 | 6·00 | 269 | 6·34 | 282 | 6·64 | 292 | 6·88 | 300 | 7·08 | 307 | 7·24 | 315 | 7·43 |
| 3·0 | 140 | 3·30 | 187 | 4·41 | 213 | 5·02 | 233 | 5·50 | 248 | 5·85 | 260 | 6·12 | 270 | 6·39 | 278 | 6·56 | 286 | 6·74 | 293 | 6·90 |
| 4·0 | 121 | 2·85 | 169 | 4·00 | 196 | 4·62 | 216 | 5·10 | 231 | 5·45 | 243 | 5·72 | 253 | 6·00 | 262 | 6·18 | 270 | 6·37 | 278 | 6·56 |
| 5·0 | 107 | 2·52 | 156 | 3·68 | 184 | 4·34 | 204 | 4·81 | 219 | 5·16 | 231 | 5·45 | 241 | 5·69 | 250 | 5·90 | 258 | 6·09 | 266 | 6·27 |
| 6·0 | 97·1 | 2·29 | 146 | 3·44 | 174 | 4·10 | 193 | 4·55 | 210 | 4·94 | 222 | 5·24 | 232 | 5·47 | 241 | 5·67 | 250 | 5·90 | 257 | 6·06 |
| 7·0 | 88·2 | 2·08 | 137 | 3·22 | 165 | 3·89 | 185 | 4·36 | 201 | 4·73 | 213 | 5·02 | 223 | 5·26 | 232 | 5·47 | 242 | 5·70 | 249 | 5·87 |
| 8·0 | 80·2 | 1·89 | 128 | 3·00 | 157 | 3·70 | 177 | 4·17 | 193 | 4·55 | 206 | 4·85 | 216 | 5·10 | 225 | 5·31 | 234 | 5·52 | 242 | 5·71 |
| 9·0 | 72·8 | 1·72 | 121 | 2·84 | 150 | 3·54 | 170 | 4·00 | 186 | 4·39 | 199 | 4·69 | 209 | 4·94 | 219 | 5·16 | 228 | 5·37 | 235 | 5·54 |
| 10 | 65·8 | 1·55 | 114 | 2·70 | 144 | 3·40 | 164 | 3·87 | 179 | 4·22 | 192 | 4·52 | 202 | 4·77 | 213 | 5·01 | 221 | 5·21 | 229 | 5·40 |
| 11 | 59·2 | 1·40 | 108 | 2·55 | 138 | 3·27 | 158 | 3·73 | 173 | 4·08 | 186 | 4·39 | 196 | 4·62 | 206 | 4·87 | 215 | 5·07 | 223 | 5·26 |
| 12 | 53·0 | 1·23 | 103 | 2·43 | 132 | 3·11 | 152 | 3·56 | 168 | 3·96 | 181 | 4·26 | 190 | 4·48 | 200 | 4·73 | 210 | 4·95 | 218 | 5·12 |
| 13 | 47·3 | 1·12 | 97·2 | 2·29 | 126 | 2·97 | 147 | 3·46 | 163 | 3·85 | 175 | 4·13 | 185 | 4·36 | 195 | 4·61 | 205 | 4·83 | 213 | 5·02 |
| 14 | 41·9 | 0·99 | 92·3 | 2·16 | 121 | 2·85 | 141 | 3·32 | 158 | 3·73 | 170 | 4·01 | 181 | 4·27 | 191 | 4·50 | 200 | 4·72 | 208 | 4·90 |
| 15 | 36·9 | 0·87 | 87·7 | 2·07 | 117 | 2·75 | 137 | 3·23 | 153 | 3·61 | 165 | 3·89 | 177 | 4·16 | 186 | 4·40 | 195 | 4·60 | 203 | 4·79 |
| 20 | 14·6 | 0·34 | 68·2 | 1·61 | 97·7 | 2·30 | 118 | 2·80 | 135 | 3·19 | 147 | 3·44 | 159 | 3·75 | 169 | 4·00 | 178 | 4·20 | 186 | 4·40 |
| 25 | .. | .. | 52·0 | 1·23 | 81·9 | 1·94 | 103 | 2·43 | 120 | 2·83 | 134 | 3·16 | 144 | 3·40 | 154 | 3·63 | 163 | 3·85 | 172 | 4·06 |
| 30 | .. | .. | 38·2 | 0·90 | 68·8 | 1·62 | 91·3 | 2·15 | 108 | 2·55 | 121 | 2·85 | 132 | 3·11 | 142 | 3·35 | 152 | 3·59 | 160 | 3·78 |
| 35 | .. | .. | 27·0 | 0·64 | 58·6 | 1·33 | 80·7 | 1·90 | 97·5 | 2·30 | 111 | 2·62 | 122 | 2·88 | 132 | 3·11 | 141 | 3·32 | 149 | 3·51 |
| 40 | .. | .. | 16·3 | 0·40 | 49·2 | 1·16 | 72·0 | 1·70 | 88·0 | 2·08 | 101 | 2·39 | 113 | 2·66 | 123 | 2·90 | 132 | 3·11 | 140 | 3·30 |
| 45 | .. | .. | 8·1 | 0·19 | 41·0 | 0·97 | 63·5 | 1·50 | 79·0 | 1·86 | 92·2 | 2·18 | 104 | 2·45 | 114 | 2·69 | 123 | 2·90 | 131 | 3·03 |
| 50 | .. | .. | .. | .. | 32·7 | 0·77 | 54·0 | 1·27 | 70·4 | 1·66 | 84·8 | 2·00 | 96·6 | 2·28 | 107 | 2·52 | 115 | 2·71 | 123 | 2·90 |
| 55 | .. | .. | .. | .. | 25·3 | 0·60 | 46·8 | 1·10 | 63·3 | 1·49 | 77·5 | 1·83 | 89·0 | 2·10 | 99·0 | 2·33 | 108 | 2·55 | 116 | 2·74 |
| 60 | .. | .. | .. | .. | 18·2 | 0·43 | 39·8 | 0·94 | 56·8 | 1·33 | 72·0 | 1·70 | 83·5 | 1·97 | 93·0 | 2·19 | 102 | 2·40 | 110 | 2·60 |
| 65 | .. | .. | .. | .. | 12·0 | 0·28 | 33·2 | 0·80 | 51·4 | 1·18 | 67·0 | 1·59 | 78·0 | 1·84 | 87·5 | 2·06 | 95·4 | 2·25 | 104 | 2·45 |
| 70 | .. | .. | .. | .. | 6·0 | 0·14 | 26·9 | 0·63 | 45·4 | 1·07 | 62·0 | 1·46 | 73·0 | 1·72 | 82·3 | 1·94 | 90·3 | 2·08 | 98·3 | 2·32 |
| 75 | .. | .. | .. | .. | .. | .. | 21·0 | 0·50 | 41·5 | 0·97 | 56·8 | 1·32 | 67·2 | 1·58 | 77·1 | 1·82 | 85·0 | 2·00 | 93·6 | 2·21 |
| 80 | .. | .. | .. | .. | .. | .. | 15·5 | 0·37 | 36·8 | 0·87 | 50·4 | 1·18 | 62·1 | 1·47 | 72·2 | 1·70 | 80·5 | 1·90 | 87·3 | 2·06 |
| 85 | .. | .. | .. | .. | .. | .. | 10·8 | 0·25 | 32·6 | 0·77 | 45·4 | 1·07 | 57·4 | 1·35 | 67·8 | 1·59 | 76·2 | 1·80 | 83·2 | 1·96 |
| 90 | .. | .. | .. | .. | .. | .. | 6·48 | 0·15 | 28·3 | 0·67 | 41·9 | 0·99 | 53·4 | 1·25 | 63·0 | 1·49 | 71·8 | 1·70 | 78·8 | 1·86 |
| 95 | .. | .. | .. | .. | .. | .. | 2·79 | 0·06 | 24·0 | 0·57 | 37·3 | 0·88 | 49·5 | 1·17 | 59·1 | 1·40 | 67·9 | 1·60 | 74·3 | 1·76 |
| 100 | .. | .. | .. | .. | .. | .. | .. | .. | 20·0 | 0·47 | 34·0 | 0·80 | 45·3 | 1·07 | 55·0 | 1·30 | 63·6 | 1·50 | 70·1 | 1·66 |

### Efficiency as affected by Jacketing.

Careful investigations show that all types of steam engines are rendered more efficient by the addition of steam jackets, and that the more completely the hot surfaces of cylinders, receivers, &c., are jacketed, the greater is the saving effected.

This amounts to saying, that for every pound of steam condensed in the jackets, some greater quantity is saved in the cylinders. The ratio of steam saved in the cylinders to steam expended in the jackets, varies from a little under 2 to 1 in some types of engine, to over 5 to 1 in other types.

The gain that may be expected to result from jacketing an engine depends on such a multitude of considerations—relating not only to the design of the engine and of the boiler, but also to the management of the machinery under steam—that it can only be very generally stated as lying between 5 and 25 per cent. of the total feed-water evaporated; but in the case of modern marine machinery, of good construction, it is not probable that the gain is over 10 per cent.

The limit of usefulness of jacketing is reached when the exhaust is just free from particles of water in suspension.

Small cylinders are more benefited by jacketing than large ones, in consequence of the ratio of area of hot surface to cubic contents being greater than in large ones; and slow moving engines benefit more than quick.

It is important that there should be a thorough circulation of steam in the jackets, but the plan of passing the steam through the jackets on its way to the H.P. cylinder should be avoided.

## THE RESISTANCE AND PROPULSION OF SHIPS.

In dealing with cubes, or with parallelepipeds of similar form, immersed in water until the uppermost face is just flush with the surface, it is found, on making the necessary calculations, that the wetted surface is exactly proportional to the $\frac{2}{3}$rd power (or the square of the cube root) of the displacement. Taking the case of cubes,—

Let $L$ = length of edge,
$D$ = displacement,
$W$ = wetted surface;

$$\text{then } D = L^3, \text{ or } L = \sqrt[3]{D},$$

$$\text{and } W = 5 \times L^2 = 5 \times (\sqrt[3]{D})^2.$$

That is, $W$ varies as $D^{\frac{2}{3}}$.

It may also be noted that $L^2 = (\sqrt[3]{D})^2$,—that is, $L^2$ (which corresponds to mid-ship section) also varies as $D^{\frac{2}{3}}$; and therefore $W$ varies as $L^2$, or, in other words, wetted surface varies as area of mid-ship section.

These results are not quite accurate for parallelepipeds which are not similar in form (that is, whose lengths, breadths, and depths are not of the same relative proportions), but the inaccuracy is only slight within practicable limits; so that, if ships of ordinary proportions are substituted for cubes or parallelepipeds, it is practically correct to say that wetted surface varies as area of mid-ship section, and also as $\frac{2}{3}$rd power of displacement.

For similar vessels displacement is also a measure of the fineness of the lines, since, when length and mid-ship section are the same, it varies directly as the prismatic co-efficient of fineness.

Now the resistances of ships depend almost entirely upon these two elements,—wetted surface, and form, or fineness of lines, and may be classed under the four heads—

(1) Resistance due to skin friction ;
(2) Resistance due to eddy making ;
(3) Resistance due to wave making ;
(4) Resistance (augmented) due to the action of the propeller on the ship.

The first of these depends on the extent and nature of the wetted surface, and the depth of immersion ; the third, on the lines of the ship, and on her degree of fineness ; and the second, on all of these combined. The fourth arises from the displacement of the water and the consequent reduction of "head" and pressure on the stern. Propellers of large diameter and fine pitch produce much *augmented resistance* as well as negative slip due to the big *wake currents*.

**Residual Resistance**—that is, the resistance from all other causes than skin friction can be estimated with a fair degree of accuracy by the following formula of Mr. D. W. Taylor, U.S.A. :—

**Rule 21.** $$\text{Residual resistance in lbs.} = \frac{12{\cdot}5\, b\mathrm{DV}^4}{\mathrm{L}^2}$$

$b$ is the *block* co-efficient ; D, the displacement in tons ; V, the speed in knots ; L, the length on waterline in feet.

This formula is applicable only to speeds for which $\frac{\mathrm{V}^2}{\mathrm{L}}$ is less than 1·2.

For merchant ships, with block co-efficients 0·6 to 0·65, this is a good formula for speeds, such that $\frac{\mathrm{V}^2}{\mathrm{L}} = 0{\cdot}85$ ; when co-efficients are 0·5 to 0·55, then $\frac{\mathrm{V}^2}{\mathrm{L}} = 1$.

It is therefore evident, from the above considerations, that the old speed and power formulæ rest on a sound basis, and are capable, in careful hands, of giving fairly accurate results. These formulæ are :—

**Rule 22.** $$\text{I.H.P.} = \frac{\mathrm{D}^{\frac{2}{3}} \times \mathrm{S}^3}{\mathrm{C}}$$

$$\text{and I.H.P.} = \frac{\text{area of immersed mid-ship section} \times \mathrm{S}^3}{\mathrm{K}}$$

where D, is displacement in tons ; S, speed in knots ; and C and K co-efficients.

The peculiar value of these formulæ lies in the fact that they can be applied at a very early stage of the work, before such data as angles of obliquity of stream lines can be obtained with any degree of accuracy: and thus the power, approximate weight, and an outline drawing of the machinery can be got out simultaneously with the design of the vessel,—a great advantage where time is limited, as is usually the case in preparing tenders.

It is important to notice, in connection with the above formulæ, that—although the resistance of a ship, moving uniformly at any speed, may vary as the *square* of that speed,—the power required to overcome the resistance, and propel it at any speed, varies as the *cube* of that speed. For,—

Let $S$ = speed in feet per minute;
$R$ = resistance in pounds at that speed;
Then $R = S^2 \times C$,—where $C$ is a co-efficient;

and, multiplying both sides by $S$

$$R \times S = S^3 \times C.$$

But $R \times S$ is the work done, in foot-pounds per minute, in overcoming the resistance $R$ through the space $S$, and, divided by 33,000, is equal to the nett horse-power required to drive the ship. This law only holds for similar ships driven at corresponding speeds (speeds proportional to the square roots of the linear dimensions), since it is evident that the resistance due to wave making can only be proportional under these conditions.

The second formula is useful as a check, or corrective to the first, where the vessels under consideration are not absolutely similar, but have same ratio of length to breadth and draught, with a variation in the rise of floor.

Table VIII. (given by Sir W. H. White) shows the values of C for some typical ships of very different classes, at various speeds, and whilst indicating generally the range of the variations that occur, serves also to show the difficulties that the naval architect has to encounter in obtaining high speeds in vessels of small dimensions.

It is, perhaps, scarcely necessary to add that these co-efficients of performance represent the *combined* efficiency of ships and machinery, and that they are therefore just as liable to be effected by an unsuitable propeller as by a foul bottom or unsuitable lines; and also that for every model there is, as a rule, only one speed of maximum efficiency, though it is evident from the type of curve usually obtained, that there may be two speeds at which the efficiency is equal and slightly below the maximum.

It should also be borne in mind that accurately determined co-efficients of performance, &c., generally apply to more or less smooth-water conditions, and that a form of vessel which gives highest speed with least power, on such trials, may be far from the best for an ocean-going steamer.

As a rule, length assists speed especially in a sea-way.

The gain in economy of propulsion resulting from increase in dimensions is made very clear by the following figures, which are derived from the trials of certain cruisers:—

Table VIII.—Relation of Powers and Speeds.

| Description of Vessel. | Length in feet. | Breadth in feet. | Draught (mean). | | Displacement in tons. | I.H.P's. and Admiralty Co-efficients. | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | 10 Knots. | | 14 Knots. | | 18 Knots. | | 20 Knots. | |
| | | | ft. | in. | | I.H.P. | Co-eff. | I.H.P. | Co-eff | I.H.P. | Co-eff. | I.H.P. | Co-eff. |
| Torpedo Boat, | 135 | 14 | 5 | 1 | 103 | 110 | 200 | 260 | 232 | 870 | 147 | 1,130 | 156 |
| Torpedo Gunboat, "Sharpshooter" Class, | 230 | 27 | 8 | 3 | 735 | 450 | 181 | 1100 | 203 | 2,500 | 190 | 3,500 | 186 |
| 3rd Class Cruiser, "Medusa," | 265 | 41 | 16 | 6 | 2,800 | 700 | 284 | 2100 | 259 | 6,400 | 181 | 10,000 | 159 |
| 2nd Class Cruiser, "Terpsichore," | 300 | 43 | 16 | 2 | 3,330 | 800 | 279 | 2400 | 255 | 6,000 | 217 | 9,000 | 198 |
| 1st Class Cruiser, "Edgar," | 360 | 60 | 23 | 9 | 7,390 | 1000 | 380 | 3000 | 347 | 7,500 | 295 | 11,000 | 276 |
| 1st Class Cruiser, "Blenheim," | 375 | 65 | 25 | 9 | 9,100 | 1500 | 290 | 4000 | 298 | 9,000 | 282 | 12,500 | 278 |
| Atlantic Passenger Steamer, | 525 | 63 | 21 | 3 | 11,550 | 2000 | 255 | 4600 | 304 | 10,000 | 297 | 14,500 | 281 |

### Table VIIIa.—Relation of Powers and Displacements.*

| | No. 1. | No. 2. | No. 3. | No. 4. | No. 5. |
|---|---|---|---|---|---|
| Length in ft. . . | 280 | 300 | 360 | 435 | 500 |
| Breadth in ft. . . | 35 | 43 | 60 | 69 | 71 |
| Mean draught in ft. . | 13 | 16½ | 23¾ | 24½ | 26¼ |
| Displacement in tons . | 1800 | 3400 | 7400 | 11000 | 14200 |
| I.H.P. for 20 knots . | 6000 | 9000 | 11000 | 14000 | 15500 |
| I.H.P. per ton of displacement . . . | 3·3 | 2·65 | 1·48 | 1·27 | 1·09 |

The following horse-powers were required to drive cruisers Nos. 4 and 5 in the above Table at the speeds named :—

| | No. 4. | No. 5. |
|---|---|---|
| 10 knots | 1500 I.H.P. | 1800 I.H.P. |
| 12 ,, | 2500 ,, | 3100 ,, |
| 14 ,, | 4000 ,, | 5000 ,, |
| 16 ,, | 6000 ,, | 7500 ,, |
| 18 ,, | 9000 ,, | 11000 ,, |
| 20 ,, | 14000 ,, | 15500 ,, |
| 22 ,, | 23000 ,, | 23000 ,, |

The frictional resistance of clean painted surfaces varies about as the 1·83 power of the speed (compare Table XVI.), but resistance due to wave making may vary very widely, since it is dependent on form. The total resistance of "Destroyers" has been found to vary as follows * :—

Up to 11 knots, . . . . nearly as speed $^{2}$
At 16 ,, . . . . ,, speed $^{3}$
,, 18–20 ,, . . . . ,, speed $^{3\cdot3}$
,, 22 ,, . . . . ,, speed $^{2\cdot7}$
,, 25 ,, . . . . ,, speed $^{2}$
,, 25–30 ,, . . . practically as speed $^{1\cdot83}$

and the resistances other than frictional vary as follows :—

Up to 11 knots as speed $^{2}$
At 12½ to 13 knots ,, speed $^{3}$
,, 14½ knots ,, speed $^{4}$
,, 18 ,, ,, speed (more than 5th power).
,, 24 ,, ,, speed $^{2}$

and at higher speeds as still lower powers of the speeds.

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* Sir Wm. White, British Association Address, 1899.

The relation of the frictional resistance to the total resistance at various speeds is computed to be as follows*:—

| | "Destroyer." | Cruiser. |
|---|---|---|
| At 12 knots | 80 per cent. | 90 per cent. |
| ,, 16 ,, | 70 ,, | 85 ,, |
| ,, 20 ,, | nearly 50 ,, | nearly 80 ,, |
| ,, 23 ,, | ... | over 70 ,, |
| ,, 30 ,, | 45 ,, | ... |

And it may be remarked that, if the coefficient of friction be doubled (as Table XVI. shows it might easily be with a foul bottom) the maximum speed of the "Destroyer" would fall fully 5 knots, and that of the cruiser would be reduced from 23 knots to 19.

Recent "Destroyers" for the British Navy have been of following dimensions, etc.:—

| | Ft. long. | Tons displ. | I.H.P. | Knots. |
|---|---|---|---|---|
| 1893 | 180 | 240 | 4000 | 26–27 |
| 1896 | 200–210 | 280– 300 | 5500–6000 | 30 |
| 1899 | 230 | 360– 380 | 8000–10,000 | 32–33 |
| 1903 | 225–230 | 550– 590 | 7000 | 25·5 |
| 1907 | 250–270 | 860– 960 | 14,000 | 33 |
| 1910 | 270–280 | 900–1000 | 12,000–16,000 | 27–33·5 |

The propelling apparatus of these boats has given, on an average, 45 I.H.P. per ton of weight. In the 30 knot boats nearly 50 per cent. of the displacement has been allotted to the propelling apparatus, and the load of fuel and equipment has taken 12 to 14 per cent. more.

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* Sir Wm. White, British Association Address, 1899.

## Table IX.—Results of Trials of Steamships.

| DESIGNATION. | TSS B—T | TSS HMSE | TSS Pg—E | TSS PE—D | TSS W—F | SS V—A | TSS F—B | TSS D—B | SS B—L | SS L—H | TSS W—D | SS B—A | SS F—A |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Length, perpendiculars, ft., | 685·0 | 360·0 | 290·0 | 270·0 | 680·0 | 466·0 | 220·0 | 530·0 | 400·0 | 240·0 | 470·0 | 320·0 | 250·0 |
| Breadth, extreme, ft., | 68·0 | 60·0 | 38·0 | 33·0 | 75·0 | 52·0 | 28·0 | 59·0 | 45·2 | 32·0 | 56·0 | 40·0 | 32·0 |
| Draught of water, mean, ft., | 29·9 | 20·6 | 11·9 | 8·8 | 30·0 | 20·1 | 9·0 | 23·5 | 18·6 | 15·4 | 24·5 | 18·3 | 12·6 |
| Displacement (tons), | 25,910 | 6100 | 2100 | 1237 | 30,020 | 8124 | 736 | 14,206 | 5842 | 2040 | 13,895 | 4720 | 1740 |
| Midship section immersed, area, | 1922 | 1059 | 416 | 266 | 2094 | 980 | 204 | 1225 | 747 | 450 | 1315 | 664 | 355 |
| Kirk's Analysis. Wetted skin, sq. ft., | 72,208 | 25,545 | 13,170 | 9678 | 76,440 | 28,511 | 6841 | 46,391 | 26,063 | 12,022 | 43,389 | 21,102 | 11,184 |
| Kirk's Analysis. Length of fore-body, | 213·0 | 158·0 | 113·3 | 106·3 | 178·0 | 176·0 | 94·0 | 124·0 | 126·3 | 81·3 | 99·0 | 71·0 | 79·0 |
| Kirk's Analysis. Angle of entrance, | 17° 4′ | 15° 56′ | 17° 33′ | 16° 5′ | 22° 11′ | 15° 10′ | 14° 12′ | 22° 47′ | 18° 2′ | 20° 0′ | 30° 20′ | 28° 38′ | 21° 36′ |
| $0{\cdot}39\sqrt[4]{L}\div\sqrt[3]{S}$., | ·725 | ·616 | ·590 | ·593 | ·787 | ·711 | ·590 | ·764 | ·711 | ·649 | ·792 | ·734 | ·678 |
| $\frac{\text{Displacement} \times 35}{\text{Length} \times \text{mid section}}$, | ·690 | ·561 | ·609 | ·605 | ·738 | ·622 | ·573 | ·765 | ·684 | ·661 | ·789 | ·771 | ·684 |
| Speed (knots), | 20·8 | 20·96 | 20·34 | 18·9 | 16·20 | 16·52 | 16·5 | 14·7 | 14·79 | 13·23 | 12·00 | 11·79 | 11·94 |
| Ind. horse-power, | 26,500 | 10,646 | 5820 | 3231 | 12,491 | 6347 | 1405 | 6118 | 3742 | 1584 | 3780 | 1788 | 962 |
| I.H.P. per 100 feet wetted skin, | 36·7 | 41·6 | 44·2 | 33·4 | 16·35 | 22·26 | 20·4 | 13·17 | 14·36 | 13·1 | 8·71 | 8·47 | 8·60 |
| Do. reduced to 10 knots, | 4·08 | 4·52 | 5·27 | 4·94 | 3·85 | 4·94 | 4·55 | 4·15 | 4·44 | 5·2 | 5·04 | 5·17 | 5·05 |
| $\frac{\text{Disp}^{\frac{2}{3}} \times S^3}{\text{I.H.P.}}$, | 297 | 288 | 236 | 246 | 329 | 287 | 261 | 305 | 280 | 240 | 264 | 258 | 255 |
| $\frac{\text{Immrsd mid sec} \times S^3}{\text{I.H.P.}}$, | 652 | 915 | 599 | 556 | 712 | 696 | 654 | 637 | 645 | 685 | 601 | 610 | 627 |
| I.H.P. ÷ displacement, | 1·02 | 1·74 | 2·77 | 2·61 | 0·42 | 0·78 | 1·90 | 0·43 | 0·65 | 0·94 | 0·27 | 0·31 | 0·55 |

## Table X.—Results of Trials of

| Particulars. | QSS LCNA | TSS KWO | TSS OCNC | TSS DLD | TSS CPA | TSS CRMA | HMS INBL | HMS DNT | HMS GLS |
|---|---|---|---|---|---|---|---|---|---|
| Length, perpdlrs, ft., | 760 | 678 | 685 | 663 | 600 | 678 | 530 | 490 | 430 |
| Breadth, extreme, ft., | 87·5 | 72·0 | 68·0 | 67·0 | 65·2 | 72·4 | 78·5 | 82·0 | 47·0 |
| Drft water, mean, ft., | 32·5 | 29·5 | 29·9 | 29·0 | 29·0 | 30·0 | 26·0 | 26·5 | 15·25 |
| Displacement (tons), | 36,440 | 26,500 | 25,910 | 23,620 | 21,500 | 27,500 | 17,250 | 17,900 | 4,800 |
| Imrsd mid sectn, sq.ft. | 2,500 | 1,950 | 1,922 | 1,753 | 1,720 | 2,064 | 1,868 | 1,912 | 624 |
| Wetted skin, sq. ft., | 82,350 | 65,150 | 67,209 | 62,110 | 54,780 | 68,700 | 47,340 | 44,630 | 22,250 |
| Prism co-efficient, . | 0·671 | 0·713 | 0·690 | 0·715 | 0·729 | 0·679 | 0·660 | 0·669 | 0·625 |
| $0{\cdot}4\sqrt[4]{L}\div\sqrt[3]{S}$., . . | 0·714 | 0·712 | 0·725 | 0·701 | 0·704 | 0·748 | 0·637 | 0·680 | 0·613 |
| Speed (knots), . . | 25·40 | 23·50 | 20·80 | 23·50 | 22·09 | 20·19 | 27·36 | 21·25 | 26·30 |
| Number of screws, . | 4 | 2 | 2 | 2 | 2 | 3 | 4 | 4 | 4 |
| Engines, . . . | Turbs | Recip | Recip | Recip | Recip | Turbs | Turbs | Turbs | Turbs |
| I.H.P. or S.H.P., . | 64,600 | 39,000 | 26,500 | 36,000 | 29,936 | 24,000 | 47,300 | 24,712 | 25,417 |
| H.P. per 100 sq. ft. wetted skin. | 78·4 | 59·90 | 39·4 | 57·9 | 54·8 | 35·00 | 100·0 | 55·4 | 114 |
| Do. redcd to 10 kns. | 4·7 | 4·62 | 4·38 | 4·47 | 5·08 | 4·25 | 4·88 | 5·78 | 6·27 |
| $D^{\frac{2}{3}}\times S^3\div$H.P. . . | 278 | 297 | 297 | 297 | 278 | 312 | 289 | 264 | 203 |
| H.P.$\div$Dispmt$^{\frac{2}{3}}$ . . | 58·7 | 43·7 | 30·2 | 44·9 | 38·8 | 26·3 | 70·8 | 36·1 | 89·2 |
| Service of ship, . . | Atl. Ex | Atl Ex | Atl Ex | Atl Ex | Atl Ex | Atl Ex | Cruiser | Bttlsp | Cruiser |

## Various Ships of High Speed.

| HMS BLN | HMS SFT | TSS BMC | HMS SKR | PSS EMQ | PSS LMG | TSS QN | TSS PG | TSS SNA | TSS NRMA | PSS LND | PSS RS |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 385 | 345 | 375 | 360 | 360 | 330 | 310 | 290 | 284 | 290 | 338 | 310 |
| 41·5 | 34·2 | 46·0 | 40·0 | 42·3 | 40·0 | 40·0 | 38·0 | 39·0 | 36·0 | 34·75 | 33 0 |
| 13·5 | 10·5 | 13·4 | 14·25 | 13·0 | 8·75 | 10·5 | 11·92 | 12·0 | 12·0 | 9·00 | 6·0 |
| 3,360 | 1,800 | 3,353 | 2,945 | 2,940 | 1,868 | 2,080 | 2,100 | 1,990 | 1,875 | 1,700 | 1,063 |
| 476 | 301 | 590 | 500 | 466 | 322 | 375 | 416 | 420 | 390 | 292 | 188 |
| 18,100 | 12,000 | 18,830 | 16,350 | 16,190 | 13,082 | 12,200 | 13,170 | 11,950 | 11,700 | 11,800 | 9,920 |
| 0·640 | 0·609 | 0·530 | 0·572 | 0·623 | 0·615 | 0·625 | 0·609 | 0·554 | 0·580 | 0·607 | 0·635 |
| 0·585 | 0·524 | 0·617 | 0·594 | 0·620 | 0·604 | 0·602 | 0·605 | 0·608 | 0·612 | 0·617 | 0·623 |
| 27·8 | 36·00 | 23·27 | 25·20 | 21·71 | 22·3 | 21·73 | 20·34 | 20·00 | 19·70 | 21·50 | 19·6 |
| 4 | 4 | 3 | 2 | Paddle | Paddle | 3 | 2 | 3 | 2 | Paddle | Pdle |
| Turbs | Turbs | Turbs | Recip | 3 Cyl Comp | Diag Comp | Turbs | 4 Ck Trpl | Turbs | Geared Turbs | Comp | Diag Comp |
| 18,000 | 30,000 | 14,700 | 16,899 | 11,442 | 7,500 | 8,000 | 5,820 | 6,670 | 5,000 | 7,000 | 3,500 |
| 99·4 | 250 | 78·1 | 103·4 | 70·6 | 57·3 | 65·6 | 44·2 | 55·8 | 42·8 | 50·3 | 35·3 |
| 4·63 | 5·35 | 6·17 | 6·46 | 6·96 | 5·16 | 6·19 | 5·27 | 6·97 | 5·60 | 5·97 | 4·69 |
| 267 | 230 | 192 | 202 | 183 | 223 | 210 | 236 | 190 | 232 | 202 | 223 |
| 80·4 | 203 | 65·6 | 82·2 | 55·8 | 49·3 | 49·1 | .. | 42·2 | 32·9 | 49·3 | 33·6 |
| Cruiser | Scout | Home Ex | Scout | Home Ex | Home Ex | Home Ex | Open S Ex | Home Ex | Home Ex | Ch'nn'l | Home Ex |

## Kirk's Analysis.

The following very simple and useful method of estimating wetted skin and comparing forms of ships is generally known as "Kirk's analysis":—

A diagram, resembling fig. 5, called the "block model" is first made, the dimensions being determined as follows:—

Length AB = length of ship (from forward side of stem to after side of stern-post, at mean trial draught).

Depth KL = depth of ship from mean trial draught to top of keel (if any).

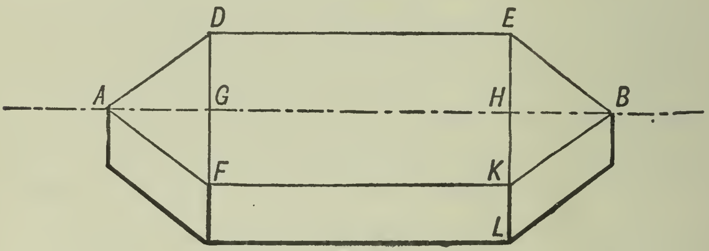

FIG. 5.

$$\text{Breadth EK} = \frac{\text{Area of immersed mid-ship section}}{\text{KL}}.$$

$$\text{AH} = \text{GB} = \frac{\text{Displacement in tons} \times 35^*}{\text{Area of immersed mid-ship section}}.$$

$$\text{Length of AD} = \sqrt{\text{AG}^2 + \text{GD}^2}.$$

Then, the wetted surface of the "block model" is,—

$$(\text{EK} \times \text{AH}) + (2\text{KL} \times \text{FK}) + (4\text{KL} \times \text{AD}).$$

The angle of entrance is EBK; EBH is half that angle; and tan. $\text{EBH} = \frac{\text{EH}}{\text{HB}}$; or, the tangent of half the angle of entrance is equal to $\frac{\text{Half breadth of model}}{\text{Length of fore-body}}$, and from this, by means of a table of natural tangents, the angle of entrance may be obtained.

---

* 35 cubic feet of salt water weigh one ton, and are therefore equal to one ton displacement.

## Table XI.—Angles of Entrance, given in degrees, suitable for various Speeds and Lengths.

| Length of Ship. | Speeds of Ship in Knots. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
| feet | | | | | | | | | | | | | | | | |
| 200 | 26·3 | 23·9 | 21·9 | 20·1 | 18·8 | 17·6 | 16·5 | 15·5 | 14·6 | 13·9 | 13·2 | 12·5 | 12·0 | 11·4 | 11·0 | 10·5 |
| 250 | 27·8 | 25·3 | 23·2 | 21·4 | 19·9 | 18·6 | 17·4 | 16·4 | 15·5 | 14·6 | 13·9 | 13·3 | 12·7 | 12·1 | 11·6 | 11·1 |
| 300 | 29·1 | 26·5 | 24·3 | 22·4 | 20·8 | 19·4 | 18·2 | 17·1 | 16·2 | 15·3 | 14·6 | 13·9 | 13·2 | 12·7 | 12·1 | 11·7 |
| 350 | 30·3 | 27·5 | 25·2 | 23·3 | 21·6 | 20·2 | 18·9 | 17·8 | 16·8 | 15·9 | 15·1 | 14·4 | 13·8 | 13·2 | 12·6 | 12·1 |
| 400 | 31·3 | 28·5 | 26·1 | 24·1 | 22·4 | 20·9 | 19·6 | 18·4 | 17·4 | 16·5 | 15·7 | 15·0 | 14·2 | 13·6 | 13·1 | 12·5 |
| 450 | 32·2 | 29·3 | 26·9 | 24·8 | 23·2 | 21·5 | 20·2 | 19·0 | 17·9 | 17·0 | 16·1 | 15·4 | 14·7 | 14·0 | 13·4 | 12·9 |
| 500 | 33·1 | 30·1 | 27·6 | 25·5 | 23·7 | 22·1 | 20·7 | 19·5 | 18·4 | 17·4 | 16·6 | 15·8 | 15·1 | 14·4 | 13·8 | 13·3 |
| 550 | 33·9 | 30·8 | 28·2 | 26·1 | 24·2 | 22·6 | 21·2 | 19·9 | 18·8 | 17·8 | 17·0 | 16·1 | 15·4 | 14·7 | 14·1 | 13·6 |
| 600 | 34·7 | 31·5 | 28·9 | 26·7 | 24·8 | 23·1 | 21·7 | 20·4 | 19·3 | 18·3 | 17·3 | 16·5 | 15·8 | 15·1 | 14·4 | 13·9 |
| 650 | 35·4 | 32·1 | 29·4 | 27·2 | 25·2 | 23·6 | 22·1 | 20·8 | 19·6 | 18·6 | 17·7 | 16·8 | 16·1 | 15·4 | 14·7 | 14·1 |
| 700 | 36·0 | 32·7 | 30·0 | 27·7 | 25·7 | 24·0 | 22·5 | 21·2 | 20·0 | 19·0 | 18·0 | 17·1 | 16·4 | 15·7 | 15·0 | 14·4 |
| 750 | 36·6 | 33·3 | 30·5 | 28·2 | 26·2 | 24·4 | 22·9 | 21·6 | 20·4 | 19·3 | 18·3 | 17·5 | 16·7 | 15·9 | 15·3 | 14·7 |
| 800 | 37·2 | 33·8 | 31·0 | 28·6 | 26·6 | 24·8 | 23·2 | 21·9 | 20·7 | 19·6 | 18·6 | 17·7 | 16·9 | 16·2 | 15·5 | 14·9 |

If ships are of very fine form for the speed required, and assuming that I.H.P. varies at $S^3$ the power per 100 feet of wetted skin at 10 knots should be 4·0 I.H.P. ; on the other hand, if the ship is not sufficiently fine for her maximum speed, the efficiency will be much less and the power at 10 knots may be then 5·0. That is, the power required to drive a ship of fairly good lines at a speed S knots is,—

**Rule 22a.** I.H.P. per 100 square feet W.S. $= \frac{S^3 \times 4 \cdot 5}{10^3} = S^3 \times 0 \cdot 0045.$

**Example** :—To find the I.H.P. necessary to drive a ship at 15 knots, the wetted skin of "block-model" being 16,200 square feet, the lines rather full.

The I.H.P. per 100 square feet $= 15^3 \times 0 \cdot 005 = 16 \cdot 875$

and I.H.P. required $= 16 \cdot 875 \times 162 = 2744.$

In ordinary practice the wetted surface of the "block model" is found to exceed that of the actual ship by 2 per cent. (in the case of full ships), by 3 to 5 per cent. for ordinary steamers, and as much as 8 per cent. in the case of very fine steamers.

The following Table gives, on inspection, the horse-powers required per 100 feet of wetted surface at various speeds and rates, and will facilitate calculations by the above method :—

## Table XII.—I.H.P. per 100 feet of wetted surface at different speeds.

| Speeds in Knots. | I.H.P. per 100 sq. feet of wetted surface at 10 knots. | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 4·0 | 4·1 | 4·2 | 4·3 | 4·4 | 4·5 | 4·6 | 4·7 | 4·8 | 4·9 | 5·0 | 5·1 | 5·2 |
| 9 | 2·92 | 2·99 | 3·06 | 3·14 | 3·21 | 3·28 | 3·35 | 3·42 | 3·49 | 3·57 | 3·64 | 3·71 | 3·78 |
| 9·5 | 3·43 | 3·51 | 3·60 | 3·68 | 3·77 | 3·85 | 3·94 | 4·02 | 4·11 | 4·19 | 4·28 | 4·86 | 4·45 |
| 10 | 4·00 | 4·10 | 4·20 | 4·30 | 4·40 | 4·50 | 4·60 | 4·70 | 4·80 | 4·90 | 5·00 | 5·10 | 5·20 |
| 10·5 | 4·63 | 4·75 | 4·86 | 4·98 | 5·09 | 5·21 | 5·32 | 5·44 | 5·56 | 5·79 | 5·79 | 5·90 | 6·02 |
| 11 | 5·32 | 5·46 | 5·59 | 5·72 | 5·86 | 5·99 | 6·12 | 6·25 | 6·39 | 6·52 | 6·65 | 6·79 | 6·92 |
| 11·5 | 6·08 | 6·23 | 6·39 | 6·54 | 6·69 | 6·84 | 6·99 | 7·15 | 7·30 | 7·45 | 7·60 | 7·76 | 7·91 |
| 12 | 6·91 | 7·08 | 7·26 | 7·43 | 7·60 | 7·78 | 7·95 | 8·12 | 8·29 | 8·47 | 8·64 | 8·81 | 8·99 |
| 12 5 | 7·81 | 8·01 | 8·20 | 8·40 | 8·59 | 8·79 | 8·98 | 9·18 | 9·37 | 9·57 | 9·76 | 9·96 | 10·15 |
| 13 | 8·79 | 9·01 | 9·23 | 9·45 | 9·67 | 9·89 | 10·10 | 10·32 | 10·54 | 10·76 | 10·98 | 11·20 | 11·42 |
| 13·5 | 9·84 | 10·09 | 10·33 | 10·58 | 10·82 | 11·07 | 11·32 | 11·56 | 11·81 | 12·05 | 12·30 | 12·55 | 12·79 |
| 14 | 10·98 | 11·25 | 11·52 | 11·80 | 12·07 | 12·35 | 12·62 | 12·90 | 13·17 | 13·44 | 13·72 | 13·99 | 14·27 |
| 14·5 | 12·19 | 12·50 | 12·80 | 13·11 | 13·41 | 13·72 | 14·02 | 14·33 | 14·63 | 14·94 | 15·24 | 15·55 | 15·85 |
| 15 | 13·50 | 13·84 | 14·17 | 14·51 | 14·85 | 15·18 | 15·52 | 15·86 | 16·19 | 16·53 | 16·87 | 17·21 | 17·54 |
| 15·5 | 14·89 | 15·27 | 15·64 | 16·01 | 16·38 | 16·76 | 17·13 | 17·50 | 17·87 | 18·25 | 18·62 | 18·99 | 19·36 |
| 16 | 16·38 | 16·79 | 17·20 | 17·61 | 18·02 | 18·43 | 18·84 | 19·25 | 19·66 | 20·07 | 20·48 | 20·89 | 21·30 |
| 16·5 | 17·97 | 18·42 | 18·87 | 19·31 | 19·76 | 20·21 | 20·66 | 21·11 | 21·56 | 22·01 | 22·46 | 22·91 | 23·36 |
| 17 | 19·65 | 20·14 | 20·63 | 21·12 | 21·62 | 22·11 | 22·60 | 23·09 | 23·58 | 24·07 | 24·56 | 25·06 | 25·55 |
| 17·5 | 21·44 | 21·97 | 22·51 | 23·04 | 23·58 | 24·12 | 24·65 | 25·19 | 25·72 | 26·26 | 26·80 | 27·33 | 27·87 |
| 18 | 23·33 | 23·91 | 24·49 | 25·08 | 25·66 | 26·24 | 26·83 | 27·41 | 27·99 | 28·58 | 29·16 | 29·74 | 30·33 |
| 18·5 | 25·33 | 25·96 | 26·59 | 27·22 | 27·86 | 28·49 | 29·12 | 29·76 | 30·39 | 31·02 | 31·66 | 32·29 | 32·92 |
| 19 | 27·44 | 28·12 | 28·81 | 29·49 | 30·18 | 30·86 | 31·55 | 32·24 | 32·92 | 33·61 | 34·29 | 34·98 | 35·67 |
| 19·5 | 29·66 | 30·40 | 31·14 | 31·88 | 32·62 | 33·37 | 34·11 | 34·85 | 35·59 | 36·33 | 37·07 | 37·81 | 38·56 |

There are several simpler methods of estimating approximately the wetted skin, such as :—

**Mumford Rule,** which is more accurate than Kirk's, is as follows :—

**Rule 23.**—Wetted skin $= (L \times D \times 1{\cdot}7) + (L \times B \times C)$

or $L(1{\cdot}7D + B.C)$.

L is the length between perpendiculars ; D is mean draught ; and B the beam, all in feet ; C is a factor which is the block co-efficient ; that is,

$$C = \frac{\text{displacement in cubic feet}}{L \times B \times D}$$

**Seaton's Rules** are,—

(*a*) **Rule 24.** Wetted skin $= (c \times d \times L) + \dfrac{D \times 35}{d}$

$c = 2$ area of immersed mid section $\div B \times d$.
For shallow draught ships $c = 2$.
For ships with high rise of floor $c = 1{\cdot}6$.
For ordinary ships whose draught of water is not less than $\frac{1}{4}$ the beam $c = 1{\cdot}8$.

D is the displacement in tons. *d* the mean moulded draught of water.

(*b*) **Rule 25.** Wetted skin $=42\sqrt[4]{K}\times D^{\frac{2}{3}}$

$$42\sqrt[4]{K}=F\;;\;K=L\div(0{\cdot}55\,B+d)$$

| Where K is | the value of F is |
|---|---|
| 4 | 59·4. |
| 5 | 62·9. |
| 6 | 65·7. |
| 8 | 70·6. |
| 9 | 72·8. |
| 10 | 74·7. |
| 11 | 76·4. |

(*c*) **Rule 26.**—Suitable prism co-efficient for speed S,—

$$P_c=0{\cdot}4\sqrt[4]{L}\div\sqrt[3]{S}.$$

(*d*) **Rule 27.**—Maximum economic speed for ship L feet long and prism co-efficient, F,—

$$S=(0{\cdot}4\sqrt[4]{L}\div P_c)^3.$$

(*e*) **Rule 28.**—Minimum length L for a speed S with a co-efficient prism F,—

$$L=\left(\frac{\sqrt[3]{S}\times P_c}{0{\cdot}4}\right)^4.$$

## Speed and Power Curves, etc.

The most reliable method of determining the I.H.P. required to drive any proposed vessel at a given speed is to base the calculations upon the results obtained from the trials of "similar" vessels; the basis of which is the fact that "similar" vessels, driven at "corresponding" speeds, have the same co-efficient of performance, when the efficiency of the machinery is the same.

"Similar" vessels are those having the same ratio of length to breadth, and to draught, and the same degree of fineness; and "corresponding" speeds are those which are proportional to the square roots of the linear dimensions of the respective vessels (*e.g.* proportional to the square root of the lengths).

Froude found that the resistance of such vessels varied almost exactly as wetted surface × $(\text{speed})^2$.

But to render the results of former trials readily accessible for such a purpose it is very desirable to have them plotted down as a series of curves, somewhat in the following manner :—

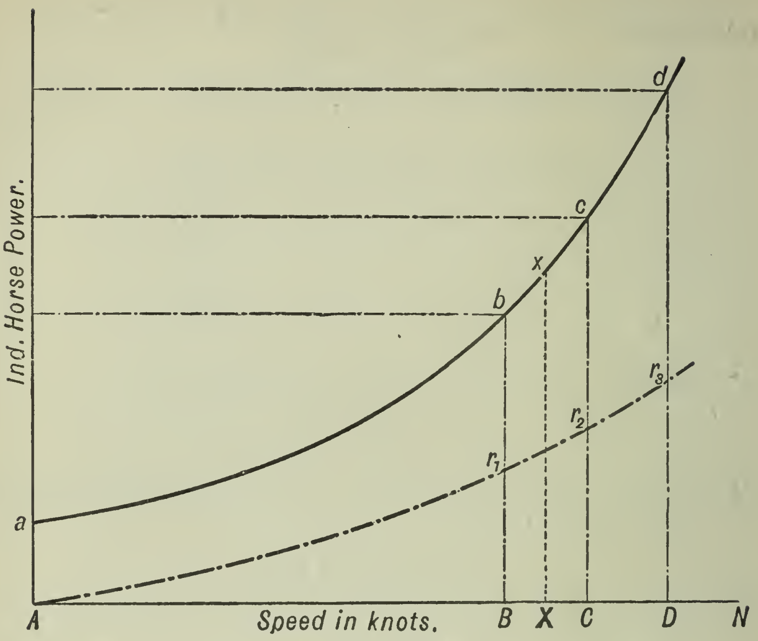

FIG. 6.

Let $P_1$, $P_2$, $P_3$ be the indicated horse-powers developed in obtaining the speeds $S_1$, $S_2$, $S_3$ knots, with $R_1$, $R_2$, $R_3$ revolutions per minute.

Take a line AN (Fig. 6) as a base line, and on it take points B, C, D, so that AB, AC, AD are proportional to $S_1$, $S_2$, $S_3$; at the points B, C, D erect ordinates B*b*, C*c*, D*d* proportional to $P_1$, $P_2$, $P_3$, and through the points *b*, *c*, and *d* draw the curve *d c b a*, which is called the "curve of power" or "curve of I.H.P." The nature of this curve is then such that if an ordinate be drawn through any other point X in the line AN, X*x* will represent the power necessary to obtain the speed AX from the same vessel, or another vessel of the same form and dimensions.

If the curve be accurately drawn, it will be found that it does not pass through A, but at a distance A*a* above that point, thus signifying that a certain amount of power is developed even at zero speed; A*a* thus represents the power required to overcome the constant friction of the engines. (See also page 17, under "Efficiency of Marine Machinery.")

Similarly a curve of revolutions may be constructed by taking points $r_1$, $r_2$, $r_3$ in the ordinates so that B*r*, C*r*, D*r* are proportional to $R_1$, $R_2$, $R_3$.

The slip may also be shown by a curve whose ordinates are proportional to the slips at the speeds $S_1$, $S_2$, $S_3$.

Examination of the curves will show:—

(1) The I.H.P., revolutions, and slip corresponding to any speed intermediate to those observed ;
(2) The constant friction, and therefore general efficiency of the engines ;
(3) The suitability of the lines of the ship for the speeds,—a sudden rise of the curve towards the higher part, showing an undue increase of resistance at the higher speeds ;
(4) The power of the speed with which the I.H.P. increases for the particular type of ship ;
(5) The suitability of the propeller to the ship,—any sudden rise in the slip curve showing that the propeller is defective, either as regards diameter or surface, or both.

Perhaps the most useful curve, however, for the purpose of determining the power required to propel some "similar" ship, of different size, at any given speed, is one constructed as shown in fig. 7, where the abscissæ represent speeds in knots, and the ordinates numerical co-efficients of performance, derived either from the Admiralty formula $\left(\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}\right)$ or other similar expression.

**Rule 28a.**—Suppose fig. 7 to represent the curve given by a vessel 250 feet long and 2400 tons displacement, and that it is required to determine the power necessary to drive a "similar" vessel of 360 feet long and 7200 tons displacement at, say, 13 knots. The question then is,—what co-efficient of performance must be assumed?—and it is answered as follows :—

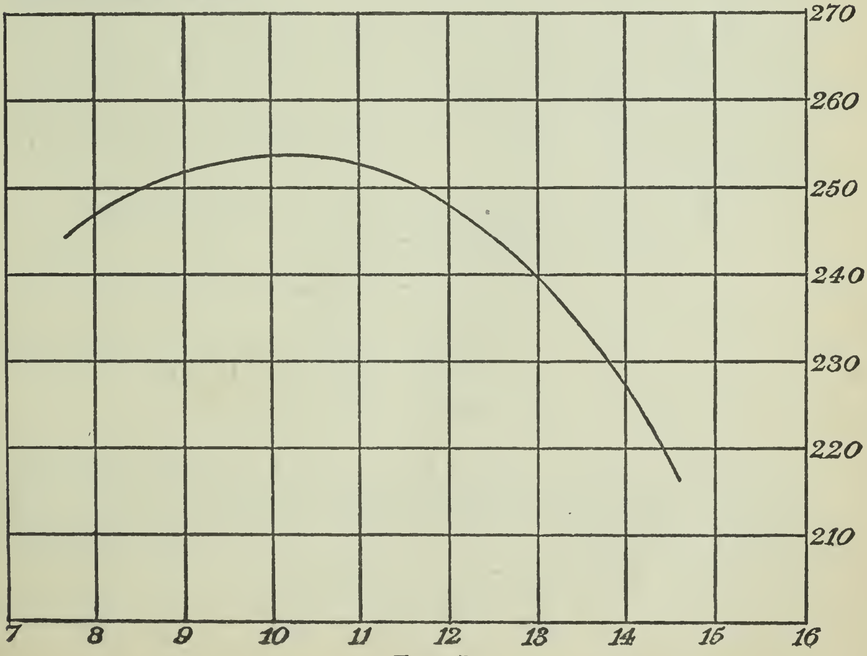

FIG. 7.

$$\sqrt{360} : \sqrt{250} :: 13 : x$$

$$\text{or } x = \frac{\sqrt{250} \times 13}{\sqrt{360}} = 10{\cdot}8 = \left\{ \begin{array}{l}\text{“corresponding” speed of} \\ \quad\text{first, or type ship,}\end{array}\right.$$

and running up the ordinate for 10·8 knots until it cuts the curve, and then along the abscissa from this point of intersection the figure 253 is found, and this, when used in connection with the formula, gives 3250 as the required I.H.P.

In cases where there are no records of exactly "similar" ships, the value of the estimate made will of course depend very largely on the experience of the estimator.

## Determination by Model Experiments.

Another method, employed by the Admiralty, and many of our leading firms of shipbuilders, for determining the power required to propel any new type of vessel is, to ascertain the resistance of a model of the new vessel in the experimental tank, and to calculate the power from the results obtained; and, where widely divergent types have to be dealt with, the method is no doubt of great value, but the expense is necessarily great.

The models are usually made of paraffin wax, from 12 feet to 20 feet long, and from $\frac{3}{4}$-inch to $1\frac{1}{4}$-inch thick; they are cast nearly to shape, and trimmed to the exact form on a shaping machine. The speeds and tension on the tow-rope are automatically recorded on paper drums driven by clock-work. The height and position of the waves created,—which are of special importance in the case of paddle vessels,—can also be noted and recorded.

The horse-power required is calculated from the resistance of the model, by the same principle of "corresponding" speeds referred to above, as follows:—

Let $l$ and $L$ = lengths of model and vessel, respectively;
$v$ and $V$ = corresponding speeds;
$r$ and $R$ = corresponding resistances;

$$\text{Then } \frac{V}{v} = \sqrt{\frac{L}{l}}$$

$$\text{and } \frac{R}{r} = \left(\frac{L}{l}\right)^3.$$

**Example.**—Suppose the E.H.P. nett or effective horse-power necessary to drive a ship 300 feet long at 15 knots is required. Let the length of the model be 12 feet, then the "corresponding" speed for it will be given by,—

$$v = 15\sqrt{\frac{12}{300}} = 3 \text{ knots.}$$

Assume the resistance of the model at this speed to be 4 lbs., then resistance of ship at 15 knots will be,—

$$R = 4 \times \left(\frac{300}{12}\right)^3 = 62,500 \text{ lbs. ;}$$

$$\text{and } \frac{62,500 \times 15 \times 6080}{60 \times 33,000} = 2879 \text{ ;}$$

or E.H.P. required is approximately 2879,—a slight correction having to be made for skin friction.

## Co-efficients of Fineness.

*The block co-efficient* expresses the ratio borne by the displacement volume to that of the parallelepiped circumscribing the immersed body.

Let $V$ = displacement in cubic feet ;
$L$ = length on water line ;
$B$ = greatest immersed breadth ;
$D$ = draught of water of body (*ex* keel) ;
$K$ = displacement co-efficient.

$$\text{Then } K = \frac{V}{L \times B \times D}.$$

*The prismatic co-efficient,*—which gives a truer measure of fineness of lines than the above,—expresses the ratio borne by the displacement volume to that of the prism swept by moving the immersed mid-ship section through the length at load water-line.

Generally speaking, the finer the water lines are, the easier is the ship driven at any speed, and consequently the speed co-efficients or multipliers vary inversely with the co-efficient of displacement. But as there is a limit to the speed at which a certain ship may be driven, any increase in power produces little or no increase in speed ; there is a limit to the fineness for the lower speeds also, so that any decrease in the co-efficient of displacement causes little or no increase in the speed co-efficients—in other words, just as the form of a ship may be inefficient for high speeds, so it may be for low.

Table XIII. gives the prismatic co-efficients appropriate to all steamers from 100 ft. to 1000 ft. long, and from 10 knots to 28 knots speed, when of the usual ship form and style.

Table XIV. shows the values of the co-efficient C in the Admiralty speed formula $\text{I.H.P.} = \frac{D^{\frac{2}{3}} \times S^3}{C}$, when the ship is in agreement with Rules 26, 27, 28.

If a ship is finer than given by Rule 26, the value of C may be increased, as it may be also when the machinery has a higher efficiency than 92 per cent.

Table XV. gives the co-efficient for computing *effective* (*nett*) horse-power necessary to overcome skin friction based on Dr. Froude's constants.

## Table XIII.—Prism Co-efficients of Ships suitable for Speeds and Lengths.

| Length in feet. | Speed in Knots per Hour. | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 10·0 | 10·5 | 11·0 | 11·5 | 12·0 | 12·5 | 13·0 | 14·0 | 15·0 | 16·0 | 17·0 | 18·0 | 19·0 | 20·0 | 21·0 | 22·0 | 23·0 | 24·0 | 25·0 | 26·0 | 27·0 | 28·0 |
| 100 | ·590 | ·578 | ·568 | ·560 | ·552 | ·545 | ·539 | ·525 | ·513 | ·501 | ·491 | ·483 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 125 | ·623 | ·611 | ·601 | ·592 | ·584 | ·576 | ·569 | ·554 | ·542 | ·531 | ·520 | ·509 | ·501 | ·492 | ·484 | .. | ... | ... | ... | ... | ... | ... |
| 150 | ·651 | ·640 | ·629 | ·620 | ·611 | ·603 | ·595 | ·580 | ·567 | ·555 | ·544 | ·534 | ·525 | ·516 | ·507 | ·499 | ·492 | ·486 | ... | ... | ... | ... |
| 175 | ·677 | ·664 | ·654 | ·645 | ·636 | ·626 | ·618 | ·603 | ·589 | ·577 | ·566 | ·554 | ·545 | ·536 | ·528 | ·520 | ·512 | ·505 | ·500 | ·492 | ·485 | ... |
| 200 | ·693 | 687 | ·678 | ·667 | ·657 | ·648 | ·640 | ·624 | ·609 | ·596 | ·584 | ·574 | ·564 | ·553 | ·544 | ·535 | ·529 | ·522 | ·515 | ·508 | ·501 | ·495 |
| 225 | ·718 | ·706 | ·696 | ·687 | ·676 | ·666 | ·658 | ·643 | ·628 | ·615 | ·602 | ·590 | ·580 | ·571 | ·562 | ·553 | 542 | ·534 | ·527 | ·520 | ·513 | ·506 |
| 250 | ·738 | ·725 | ·714 | ·702 | ·693 | ·685 | ·675 | ·659 | ·641 | ·630 | ·618 | ·607 | ·595 | ·585 | ·576 | ·568 | ·560 | ·552 | ·544 | ·537 | ·530 | ·523 |
| 275 | ·755 | ·742 | ·731 | ·720 | ·710 | ·700 | ·692 | ·674 | ·659 | ·646 | ·632 | ·620 | ·619 | ·599 | ·588 | ·579 | ·573 | ·563 | ·557 | ·550 | ·543 | ·536 |
| 300 | ·772 | ·760 | ·740 | ·737 | ·727 | ·716 | ·707 | ·690 | ·674 | ·660 | ·647 | ·634 | ·623 | ·613 | ·603 | ·594 | ·586 | ·578 | ·570 | ·563 | ·555 | ·548 |
| 325 | ·784 | ·773 | ·762 | ·752 | ·741 | ·731 | ·720 | ·703 | ·689 | ·672 | ·660 | ·648 | ·636 | ·626 | ·615 | ·606 | ·598 | ·590 | ·582 | ·574 | ·566 | ·559 |
| 350 | ·802 | ·790 | ·779 | ·769 | ·759 | ·748 | ·739 | ·720 | ·704 | ·684 | ·675 | ·663 | ·648 | ·640 | ·627 | ·618 | ·609 | ·601 | ·592 | ·584 | ·577 | ·570 |
| 375 | ·815 | ·803 | ·791 | ·780 | ·769 | ·759 | ·748 | ·730 | ·713 | ·697 | ·684 | ·671 | ·659 | ·650 | ·636 | ·627 | ·619 | ·611 | ·603 | ·595 | ·587 | ·580 |
| 400 | ·829 | ·816 | ·804 | ·792 | ·781 | ·771 | ·760 | ·743 | ·725 | ·709 | ·696 | ·681 | ·669 | ·658 | ·647 | ·638 | ·629 | ·621 | ·613 | ·604 | ·596 | ·590 |
| 450 | ·853 | ·840 | ·828 | ·816 | ·804 | ·793 | ·783 | ·764 | ·746 | ·731 | ·716 | ·702 | ·690 | ·678 | ·667 | ·656 | ·648 | ·640 | ·631 | ·622 | ·614 | ·601 |
| 500 | ·878 | ·863 | ·850 | ·837 | ·826 | ·815 | ·804 | ·784 | ·767 | ·750 | ·735 | ·721 | ·708 | ·697 | ·687 | ·677 | ·666 | ·657 | ·648 | ·640 | ·631 | ·620 |
| 550 | ·899 | ·885 | ·870 | ·858 | ·846 | ·834 | ·823 | ·805 | ·785 | ·768 | ·754 | ·739 | ·726 | ·713 | ·702 | ·693 | ·682 | ·672 | ·663 | ·654 | ·645 | ·637 |
| 600 | ·918 | ·904 | ·891 | ·877 | ·864 | ·853 | ·841 | ·821 | ·801 | ·785 | ·770 | ·756 | ·742 | ·729 | ·717 | ·707 | ·697 | ·687 | ·678 | ·670 | ·660 | ·650 |
| 650 | ·937 | ·923 | ·909 | ·896 | ·882 | ·871 | ·860 | ·838 | ·820 | ·801 | ·786 | ·771 | ·758 | ·743 | ·732 | ·720 | ·711 | ·701 | ·692 | ·682 | ·673 | ·664 |
| 700 | ·952 | ·940 | ·926 | ·910 | ·898 | ·886 | ·871 | ·855 | ·832 | ·816 | ·800 | ·785 | ·770 | ·759 | ·745 | ·734 | ·724 | ·714 | ·704 | ·695 | ·685 | ·677 |
| 750 | ... | ·955 | ·942 | ·925 | ·912 | ·901 | ·890 | ·870 | ·847 | ·830 | ·814 | ·800 | ·783 | ·772 | ·758 | ·743 | ·736 | ·726 | ·716 | ·707 | ·697 | ·688 |
| 800 | ... | ... | ·959 | ·941 | ·929 | ·917 | ·902 | ·883 | ·861 | ·844 | ·828 | ·812 | ·800 | ·785 | ·771 | ·760 | ·750 | ·739 | ·729 | ·719 | ·709 | ·700 |
| 850 | ... | ... | ... | ·956 | ·943 | ·931 | ·915 | ·900 | ·874 | ·857 | ·840 | ·824 | ·810 | ·797 | ·782 | ·771 | ·761 | ·750 | ·740 | ·729 | ·720 | ·710 |
| 900 | ... | ... | ... | ... | ·957 | ·945 | ·930 | ·910 | ·887 | ·870 | ·853 | ·837 | ·821 | ·809 | ·796 | ·783 | ·772 | ·761 | ·750 | ·740 | ·731 | ·721 |
| 950 | ... | ... | ... | ... | ... | ·957 | ·940 | ·921 | ·900 | ·881 | ·864 | ·848 | ·832 | ·819 | ·805 | ·793 | ·782 | ·771 | ·760 | ·750 | ·740 | ·730 |
| 1,000 | ... | ... | ... | ... | ... | ... | ·954 | ·934 | ·912 | ·900 | ·876 | ·860 | ·843 | ·831 | ·816 | 805 | ·793 | ·782 | ·770 | ·760 | ·750 | ·740 |

Table XIV.—Values of Co-efficient C in Formula I.H.P. $= \frac{D^{\frac{2}{3}} \times S^3}{C}$ when Co-efficient of Water Lines is not greater than $0{\cdot}4 \frac{\sqrt[4]{L}}{\sqrt[3]{S}}$.

| Speed in knots. | 10 | 12 | 14 | 16 | 18 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 100 feet long, . . | 219 | 206 | 196 | 188 | 181 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 150 ,, . . | 244 | 229 | 217 | 209 | 201 | 193 | 190 | 188 | 185 | 182 | 180 | ... | ... | ... |
| 200 ,, . . | 261 | 245 | 233 | 224 | 215 | 208 | 204 | 201 | 198 | 195 | 193 | 191 | 188 | 186 |
| 250 ,, . . | 277 | 260 | 247 | 236 | 228 | 219 | 216 | 213 | 210 | 207 | 205 | 202 | 199 | 197 |
| 300 ,, . . | 289 | 272 | 258 | 247 | 238 | 230 | 226 | 223 | 220 | 216 | 214 | 211 | 208 | 206 |
| 350 ,, . . | 301 | 282 | 269 | 257 | 247 | 240 | 235 | 232 | 228 | 225 | 222 | 219 | 216 | 213 |
| 400 ,, . . | 311 | 292 | 277 | 266 | 256 | 248 | 243 | 240 | 236 | 232 | 230 | 227 | 224 | 221 |
| 450 ,, . . | 321 | 301 | 286 | 274 | 263 | 255 | 250 | 247 | 243 | 240 | 237 | 234 | 231 | 228 |
| 500 ,, . . | 330 | 308 | 293 | 281 | 271 | 262 | 257 | 253 | 250 | 246 | 243 | 240 | 237 | 234 |
| 550 ,, . . | ... | 316 | 301 | 288 | 277 | 268 | 263 | 258 | 256 | 252 | 249 | 245 | 242 | 239 |
| 600 ,, . . | ... | 323 | 307 | 294 | 282 | 274 | 268 | 264 | 261 | 257 | 254 | 250 | 246 | 245 |
| 650 ,, . . | ... | 330 | 314 | 301 | 289 | 279 | 274 | 271 | 267 | 263 | 260 | 256 | 253 | 249 |
| 700 ,, . . | ... | ... | 320 | 306 | 294 | 284 | 280 | 276 | 272 | 268 | 265 | 261 | 258 | 254 |
| 750 ,, . . | ... | ... | 325 | 311 | 299 | 289 | 284 | 280 | 276 | 272 | 269 | 265 | 261 | 258 |
| 800 ,, . . | ... | ... | 330 | 317 | 304 | 294 | 289 | 285 | 280 | 277 | 273 | 270 | 266 | 263 |
| 850 ,, . . | ... | ... | ... | 321 | 308 | 299 | 293 | 289 | 285 | 281 | 278 | 274 | 270 | 267 |
| 900 ,, . . | ... | ... | ... | 326 | 312 | 303 | 298 | 294 | 289 | 285 | 282 | 278 | 274 | 271 |

## Immersed Surface Friction.

Table XVI. (page 47) gives a general statement of the results of Froude's experiments on this subject; they were made on boards $^3/_{16}$ in. thick and 19 in. deep, which were coated with the substances to be experimented on, and towed edgeways through the water. The resistances are given in lbs. per square foot at the standard speed of 600 feet per minute, and, as the power of the speed to which the friction is proportional is also given, the resistance at other speeds is easily calculated.*

Columns A give the power of the speed to which the resistance is approximately proportional; columns B give the mean resistance per square foot of the whole surface of a board of the lengths stated in the table; columns C give the resistance of a square foot of surface at the distance sternward from the cut-water stated in the heading.

## Table XV.—Co-efficients for Computing Effective Horse-power required to overcome Skin Friction based on Mr Froude's Constants, as given by Mr A. W. Johns.

If S is the wetted surface in square feet, then

E.H.P. $= f.\ S$, where $f$ has the values given below.

| Speed in Knots. | Length of Ship in Feet. | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 100 | 150 | 200 | 250 | 300 | 350 | 400 | 450 | 500 |
| 25, . | ·2516 | ·2477 | ·2458 | ·2444 | ·2434 | ·2415 | ·2415 | ·2407 | ·2399 |
| 24, . | ·2242 | ·2207 | ·2190 | ·2178 | ·2169 | ·2160 | ·2152 | ·2145 | ·2138 |
| 23, . | ·1988 | ·1957 | ·1942 | ·1931 | ·1923 | ·1916 | ·1908 | ·1902 | ·1895 |
| 22, . | ·1753 | ·1726 | ·1713 | ·1703 | ·1696 | ·1690 | ·1683 | ·1677 | ·1672 |
| 21, . | ·1537 | ·1514 | ·1502 | ·1494 | ·1487 | ·1481 | ·1476 | ·1471 | ·1466 |
| 20, . | ·1340 | ·1319 | ·1308 | ·1301 | ·1296 | ·1291 | ·1286 | ·1281 | ·1277 |
| 19, . | ·1159 | ·1141 | ·1132 | ·1126 | ·1121 | ·1171 | ·1112 | ·1108 | ·1105 |
| 18, . | ·0995 | ·0979 | ·0972 | ·0966 | ·0962 | ·0958 | ·0955 | ·0951 | ·0948 |
| 17, . | ·0846 | ·0833 | ·0827 | ·0822 | ·0819 | ·0815 | ·0812 | ·0810 | ·0807 |
| 16, . | ·0713 | ·0702 | ·0697 | ·0693 | ·0690 | ·0687 | ·0685 | ·0682 | ·0680 |
| 15, . | ·0594 | ·0585 | ·0580 | ·0577 | ·0575 | ·0573 | ·0570 | ·0568 | ·0567 |
| 14, . | ·0489 | ·0481 | ·0478 | ·0475 | ·0473 | ·0471 | ·0469 | ·0468 | ·0466 |
| 13, . | ·0397 | ·0390 | ·0387 | ·0385 | ·0384 | ·0382 | ·0381 | ·0379 | ·0378 |
| 12, . | ·0315 | ·0312 | ·0309 | ·0308 | ·0307 | ·0305 | ·0304 | ·0303 | ·0302 |

In the above table skin friction is taken as varying as $V^{1\cdot825}$.

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* NOTE.—See Rule 30.

## Table XVI.—Resistances of Surfaces.

| Nature of Surface. | Length of surface, or distance from cutwater, in feet. | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 Feet. | | | 8 Feet. | | | 20 Feet. | | | 50 Feet. | | |
| | A | B | C | A | B | C | A | B | C | A | B | C |
| Varnish, . | 2·00 | ·41 | ·390 | 1·85 | ·325 | ·264 | 1·85 | ·278 | ·240 | 1·83 | ·250 | ·226 |
| Paraffin, . | 1·95 | ·38 | ·370 | 1·94 | ·314 | ·260 | 1·93 | ·271 | ·237 | ... | ... | ... |
| Tinfoil, . | 2·16 | ·30 | ·295 | 1·99 | ·278 | ·263 | 1·90 | ·262 | ·244 | 1·83 | ·246 | ·232 |
| Calico, . . | 1·93 | ·87 | ·725 | 1·92 | ·626 | ·504 | 1·89 | ·531 | ·447 | 1·87 | ·474 | ·423 |
| Fine sand, | 2·00 | ·81 | ·690 | 2·00 | ·583 | ·450 | 2·00 | ·480 | ·384 | 2·06 | ·405 | ·337 |
| Medium ,, | 2·00 | ·90 | ·730 | 2·00 | ·625 | ·488 | 2·00 | ·534 | ·465 | 2·00 | ·488 | ·456 |
| Coarse ,, | 2·00 | 1·10 | ·880 | 2·00 | ·714 | ·520 | 2·00 | ·588 | ·490 | .. | ... | ... |

## True Mean Speed.

To determine the true mean speed of a vessel when the runs are taken on the measured mile, half with the tide, and half against :—

### Example.

| Runs. | Observed Speeds. | 1st Means. | 2nd Means. | 3rd Means. | 4th Means. | Mean of Means. |
|---|---|---|---|---|---|---|
| 1st | 18·5 | | | | | |
| | | 15·90 | | | | |
| 2nd | 13·3 | | 15·625 | | | |
| | | 15·35 | | 15·5750 | | |
| 3rd | 17·4 | | 15·525 | | 15·52500 | |
| | | 15·70 | | 15·4750 | | 15·478125 |
| 4th | 14·0 | | 15·425 | | 15·43125 | True mean |
| | | 15·15 | | 15·3875 | | speed. |
| 5th | 16·3 | | 15·350 | | | |
| | | 15·55 | | | | |
| 6th | 14·8 | | 4)61·925 | | | |
| | | | 15·48125 | | | |
| | 6)94·2 | | Ordinary mean of second means. | | | |
| | 15·70 | | | | | |
| | Ordinary mean speed. | | | | | |

The ordinary mean of second means is generally taken,—as unavoidable errors of observation render the third and following decimal places of very doubtful value.

Find the means of consecutive speeds continually found until only one remains.

If a, b, c, d, e, and f are the speeds as given of six runs on the measured mile, the ultimate mean of means may be found by the following rule.

**Rule 29.** $$\text{Mean speed} = \frac{(a+f)+5\,(b+e)+10\,(c+d)}{32}.$$

Taking the above example by this rule, the mean speed is 15·478 knots. It is, however, usual now when taking the speed as the ultimate mean of means to have an odd number of runs on the mile for greater accuracy in a tideway; then if five runs are made:—

$$\text{Mean Speed} = \frac{a+4b+6c+4d+e}{16}$$

Taking the first five runs in the above example, the mean speed is then 15·525.

## Relation of Speeds and Powers.

**Given two speeds of a vessel, and the corresponding horse-powers, to find what power of the speed the horse-power varies as :—**

Let $s$ and $S$ = the two speeds.
,, $p$ and $P$ = the corresponding powers.
,, $x$ = power of $s$ and $S$ that $p$ and $P$ vary as

$$\text{Then } \frac{S^x}{s^x} = \frac{P}{p}$$

$$\text{or } x\,(\text{Log } S - \text{Log } s) = \text{Log } P - \text{Log } p$$

**Rule 30.** $$\text{And } x = \frac{\text{Log } P - \text{Log } p}{\text{Log } S - \text{Log } s}.$$

## Depth of Water for Speed Trials.

Dr D. W. Taylor's formula :

D is the draught of water of ship in feet.
L is her length in feet.
S the maximum speed in knots.

**Rule 30a.** $$\text{Minimum depth} = \frac{10 \times D \times S}{L} \text{ fathoms.}$$

## Table XVII.—Speed in Knots from time of doing Nautical Mile.

### TIMES AND SPEEDS.

The number in this Table corresponding to the time in which a vessel passes over the measured mile is her speed in knots.

| Secs. | 2 min. | 3 min. | 4 min. | 5 min. | 6 min. | 7 min. | 8 min. | 9 min. | 10 min. | 11 min. | 12 min. | 13 min. | 14 min. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 30·000 | 20·000 | 15·000 | 12·000 | 10·000 | 8·571 | 7·500 | 6·667 | 6·000 | 5·455 | 5·000 | 4·615 | 4·286 |
| 1 | 29·752 | 19·890 | 14·938 | 11·960 | 9·972 | 8·551 | 7·484 | 6·654 | 5·990 | 5·446 | 4·993 | 4·609 | 4·281 |
| 2 | 29·508 | 19·780 | 14·876 | 11·921 | 9·945 | 8·531 | 7·469 | 6·642 | 5·980 | 5·438 | 4·986 | 4·604 | 4·275 |
| 3 | 29·268 | 19·672 | 14·815 | 11·881 | 9·917 | 8·511 | 7·453 | 6·630 | 5·970 | 5·430 | 4·979 | 4·598 | 4·270 |
| 4 | 29·032 | 19·565 | 14·754 | 11·842 | 9·890 | 8·491 | 7·438 | 6·618 | 5·960 | 5·422 | 4·972 | 4·592 | 4·265 |
| 5 | 28·800 | 19·459 | 14·694 | 11·803 | 9·863 | 8·471 | 7·423 | 6·606 | 5·950 | 5·414 | 4·965 | 4·586 | 4·260 |
| 6 | 28·571 | 19·355 | 14·634 | 11·765 | 9·836 | 8·451 | 7·407 | 6·593 | 5·941 | 5·405 | 4·959 | 4·580 | 4·255 |
| 7 | 28·346 | 19·251 | 14·575 | 11·726 | 9·809 | 8·431 | 7·392 | 6·581 | 5·931 | 5·397 | 4·952 | 4·574 | 4·250 |
| 8 | 28·125 | 19·149 | 14·516 | 11·688 | 9·783 | 8·411 | 7·377 | 6·569 | 5·921 | 5·389 | 4·945 | 4·568 | 4·245 |
| 9 | 27·907 | 19·048 | 14·458 | 11·650 | 9·756 | 8·392 | 7·362 | 6·557 | 5·911 | 5·381 | 4·938 | 4·563 | 4·240 |
| 10 | 27·692 | 18·947 | 14·400 | 11·613 | 9·730 | 8·372 | 7·347 | 6·545 | 5·902 | 5·373 | 4·931 | 4·557 | 4·235 |
| 11 | 27·481 | 18·848 | 14·343 | 11·576 | 9·704 | 8·353 | 7·332 | 6·534 | 5·892 | 5·365 | 4·925 | 4·551 | 4·230 |
| 12 | 27·273 | 18·750 | 14·286 | 11·538 | 9·677 | 8·333 | 7·317 | 6·522 | 5·882 | 5·357 | 4·918 | 4·545 | 4·225 |
| 13 | 27·068 | 18·653 | 14·229 | 11·502 | 9·651 | 8·314 | 7·302 | 6·510 | 5·873 | 5·349 | 4·911 | 4·540 | 4·220 |
| 14 | 26·866 | 18·557 | 14·173 | 11·465 | 9·626 | 8·295 | 7·287 | 6·498 | 5·863 | 5·341 | 4·905 | 4·534 | 4·215 |
| 15 | 26·667 | 18·461 | 14·118 | 11·429 | 9·600 | 8·276 | 7·273 | 6·486 | 5·854 | 5·333 | 4·898 | 4·528 | 4·210 |
| 16 | 26·471 | 18·367 | 14·062 | 11·392 | 9·574 | 8·257 | 7·258 | 6·475 | 5·844 | 5·325 | 4·891 | 4·523 | 4·206 |
| 17 | 26·277 | 18·274 | 14·008 | 11·356 | 9·549 | 8·238 | 7·243 | 6·463 | 5·835 | 5·318 | 4·885 | 4·517 | 4·201 |
| 18 | 26·087 | 18·182 | 13·953 | 11·321 | 9·524 | 8·219 | 7·229 | 6·452 | 5·825 | 5·310 | 4·878 | 4·511 | 4·196 |
| 19 | 25·899 | 18·090 | 13·900 | 11·285 | 9·499 | 8·200 | 7·214 | 6·440 | 5·816 | 5·302 | 4·871 | 4·506 | 4·191 |
| Secs. | 2 min. | 3 min. | 4 min. | 5 min. | 6 min. | 7 min. | 8 min. | 9 min. | 10 min. | 11 min. | 12 min. | 13 min. | 14 min. |

Should the speed be required in kilometres,—multiply the speed in knots, from the above Table, by 1·853.
See also Table CLXXX. of knots, miles, kilometres, &c.

Table XVII.—*continued.*

## TIMES AND SPEEDS.

The number in this Table corresponding to the time in which a vessel passes over the measured mile is her speed in knots.

| Secs. | 2 min. | 3 min. | 4 min. | 5 min. | 6 min. | 7 min. | 8 min. | 9 min. | 10 min. | 11 min. | 12 min. | 13 min. | 14 min. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 20 | 25·714 | 18·000 | 13·846 | 11·250 | 9·474 | 8·182 | 7·200 | 6·429 | 5·806 | 5·294 | 4·865 | 4·500 | 4·186 |
| 21 | 25·532 | 17·910 | 13·793 | 11·215 | 9·449 | 8·163 | 7·186 | 6·417 | 5·797 | 5·286 | 4·858 | 4·494 | 4·181 |
| 22 | 25·352 | 17·822 | 13·740 | 11·180 | 9·424 | 8·145 | 7·171 | 6·406 | 5·788 | 5·279 | 4·852 | 4·489 | 4·176 |
| 23 | 25·175 | 17·734 | 13·688 | 11·146 | 9·399 | 8·126 | 7·157 | 6·394 | 5·778 | 5·271 | 4·845 | 4·483 | 4·171 |
| 24 | 25·000 | 17·647 | 13·636 | 11·111 | 9·375 | 8·108 | 7·143 | 6·383 | 5·769 | 5·263 | 4·839 | 4·478 | 4·167 |
| 25 | 24·828 | 17·561 | 13·585 | 11·077 | 9·351 | 8·090 | 7·129 | 6·372 | 5·760 | 5·255 | 4·832 | 4·472 | 4·162 |
| 26 | 24·658 | 17·476 | 13·534 | 11·043 | 9·326 | 8·072 | 7·115 | 6·360 | 5·751 | 5·248 | 4·826 | 4·466 | 4·157 |
| 27 | 24·490 | 17·391 | 13·483 | 11·009 | 9·302 | 8·054 | 7·101 | 6·349 | 5·742 | 5·240 | 4·819 | 4·461 | 4·152 |
| 28 | 24·324 | 17·308 | 13·433 | 10·976 | 9·278 | 8·036 | 7·087 | 6·338 | 5·732 | 5·233 | 4·813 | 4·455 | 4·147 |
| 29 | 24·161 | 17·225 | 13·383 | 10·942 | 9·254 | 8·018 | 7·073 | 6·327 | 5·723 | 5·225 | 4·806 | 4·450 | 4·143 |
| 30 | 24·000 | 17·143 | 13·333 | 10·909 | 9·231 | 8·000 | 7·059 | 6·316 | 5·714 | 5·217 | 4·800 | 4·444 | 4·138 |
| 31 | 23·841 | 17·062 | 13·284 | 10·876 | 9·207 | 7·982 | 7·045 | 6·305 | 5·705 | 5·210 | 4·794 | 4·439 | 4·133 |
| 32 | 23·684 | 16·981 | 13·235 | 10·843 | 9·184 | 7·965 | 7·031 | 6·294 | 5·696 | 5·202 | 4·787 | 4·433 | 4·128 |
| 33 | 23·529 | 16·901 | 13·187 | 10·811 | 9·160 | 7·947 | 7·018 | 6·283 | 5·687 | 5·195 | 4·781 | 4·428 | 4·124 |
| 34 | 23·377 | 16·822 | 13·139 | 10·778 | 9·137 | 7·930 | 7·004 | 6·272 | 5·678 | 5·187 | 4·774 | 4·423 | 4·119 |
| 35 | 23·226 | 16·744 | 13·091 | 10·746 | 9·114 | 7·912 | 6·990 | 6·261 | 5·669 | 5·180 | 4·768 | 4·417 | 4·114 |
| 36 | 23·077 | 16·667 | 13·043 | 10·714 | 9·091 | 7·895 | 6·977 | 6·250 | 5·660 | 5·172 | 4·762 | 4·412 | 4·110 |
| 37 | 22·930 | 16·590 | 12·996 | 10·682 | 9·068 | 7·877 | 6·963 | 6·239 | 5·651 | 5·165 | 4·756 | 4·406 | 4·105 |
| 38 | 22·785 | 16·514 | 12·950 | 10·651 | 9·045 | 7·860 | 6·950 | 6·228 | 5·643 | 5·158 | 4·749 | 4·401 | 4·100 |
| 39 | 22·642 | 16·438 | 12·903 | 10·619 | 9·023 | 7·843 | 6·936 | 6·218 | 5·634 | 5·150 | 4·743 | 4·396 | 4·096 |
| Secs. | 2 min. | 3 min. | 4 min. | 5 min. | 6 min. | 7 min. | 8 min. | 9 min. | 10 min. | 11 min. | 12 min. | 13 min. | 14 min. |

Should the speed be required in kilometres,—multiply the speed in knots, from the above Table, by 1·853.
See also Table CLXXX. of knots, miles, kilometres, &c.

## TIMES AND SPEEDS.

The number in this Table corresponding to the time in which a vessel passes over the measured mile is her speed in knots.

| Secs. | 1 min. | 2 min. | 3 min. | 4 min. | 5 min. | 6 min. | 7 min. | 8 min. | 9 min. | 10 min. | 11 min. | 12 min. | 13 min. | 14 min. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 40 | 36·000 | 22·500 | 16·364 | 12·857 | 10·538 | 9·000 | 7·826 | 6·923 | 6·207 | 5·625 | 5·143 | 4·737 | 4·390 | 4·091 |
| 41 | 35·644 | 22·360 | 16·290 | 12·811 | 10·557 | 8·978 | 7·809 | 6·910 | 6·196 | 5·616 | 5·136 | 4·731 | 4·385 | 4·086 |
| 42 | 35·294 | 22·222 | 16·216 | 12·766 | 10·526 | 8·955 | 7·792 | 6·897 | 6·186 | 5·607 | 5·128 | 4·724 | 4·379 | 4·082 |
| 43 | 34·951 | 22·086 | 16·143 | 12·721 | 10·496 | 8·933 | 7·775 | 6·883 | 6·175 | 5·599 | 5·121 | 4·718 | 4·374 | 4·077 |
| 44 | 34·615 | 21·951 | 16·071 | 12·676 | 10·465 | 8·911 | 7·759 | 6·870 | 6·164 | 5·590 | 5·114 | 4·712 | 4·369 | 4·072 |
| 45 | 34·286 | 21·818 | 16·000 | 12·632 | 10·435 | 8·889 | 7·742 | 6·857 | 6·154 | 5·581 | 5·106 | 4·706 | 4·364 | 4·068 |
| 46 | 33·962 | 21·687 | 15·929 | 12·587 | 10·405 | 8·867 | 7·725 | 6·844 | 6·143 | 5·573 | 5·099 | 4·700 | 4·358 | 4·063 |
| 47 | 33·644 | 21·557 | 15·859 | 12·544 | 10·375 | 8·845 | 7·709 | 6·831 | 6·133 | 5·564 | 5·092 | 4·693 | 4·353 | 4·059 |
| 48 | 33·333 | 21·429 | 15·789 | 12·500 | 10·345 | 8·824 | 7·692 | 6·818 | 6·122 | 5·556 | 5·085 | 4·687 | 4·348 | 4·054 |
| 49 | 33·028 | 21·302 | 15·721 | 12·457 | 10·315 | 8·802 | 7·676 | 6·805 | 6·112 | 5·547 | 5·078 | 4·681 | 4·343 | 4·049 |
| 50 | 32·727 | 21·176 | 15·652 | 12·414 | 10·286 | 8·780 | 7·660 | 6·792 | 6·102 | 5·538 | 5·070 | 4·675 | 4·337 | 4·045 |
| 51 | 32·432 | 21·053 | 15·584 | 12·371 | 10·256 | 8·759 | 7·643 | 6·780 | 6·091 | 5·530 | 5·063 | 4·669 | 4·332 | 4·040 |
| 52 | 32·143 | 20·930 | 15·517 | 12·329 | 10·227 | 8·738 | 7·627 | 6·767 | 6·081 | 5·521 | 5·056 | 4·663 | 4·327 | 4·035 |
| 53 | 31·858 | 20·809 | 15·451 | 12·287 | 10·198 | 8·717 | 7·611 | 6·754 | 6·071 | 5·513 | 5·049 | 4·657 | 4·322 | 4·031 |
| 54 | 31·579 | 20·690 | 15·385 | 12·245 | 10·169 | 8·696 | 7·595 | 6·742 | 6·061 | 5·505 | 5·042 | 4·651 | 4·316 | 4·027 |
| 55 | 31·304 | 20·571 | 15·319 | 12·203 | 10·141 | 8·675 | 7·579 | 6·729 | 6·050 | 5·496 | 5·035 | 4·645 | 4·311 | 4·022 |
| 56 | 31·034 | 20·455 | 15·254 | 12·162 | 10·112 | 8·654 | 7·563 | 6·716 | 6·040 | 5·488 | 5·028 | 4·639 | 4·306 | 4·018 |
| 57 | 30·769 | 20·339 | 15·190 | 12·121 | 10·084 | 8·633 | 7·547 | 6·704 | 6·030 | 5·479 | 5·021 | 4·633 | 4·301 | 4·013 |
| 58 | 30·508 | 20·225 | 15·126 | 12·081 | 10·056 | 8·612 | 7·531 | 6·691 | 6·020 | 5·471 | 5·014 | 4·627 | 4·296 | 4·009 |
| 59 | 30·252 | 20·112 | 15·063 | 12·040 | 10·028 | 8·592 | 7·516 | 6·679 | 6·010 | 5·463 | 5·007 | 4·621 | 4·291 | 4·004 |
| Secs. | 1 min. | 2 min. | 3 min. | 4 min. | 5 min. | 6 min. | 7 min. | 8 min. | 9 min. | 10 min. | 11 min. | 12 min. | 13 min. | 14 min. |

Should the speed be required in kilometres,—multiply the speed in knots, from the above Table, by 1·853.
See also Table CLXXX. of knots, miles, kilometres, &c.

The following Table of the ⅔ powers of numbers will be of service in all calculations for which displacement is taken as a basis.

Table XVIII.—Two-thirds powers of numbers.

| Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. |
|---|---|---|---|---|---|---|---|
| 100 | 21·54 | 480 | 61·30 | 860 | 90·43 | 1240 | 115·42 |
| 110 | 22·96 | 490 | 62·15 | 870 | 91·13 | 50 | 116·04 |
| 120 | 24·33 | 500 | 62·99 | 880 | 91·83 | 60 | 116·66 |
| 130 | 25·66 | 510 | 63·83 | 890 | 92·52 | 70 | 117·27 |
| 140 | 26·96 | 520 | 64·66 | 900 | 93·22 | 80 | 117·89 |
| 150 | 28·23 | 530 | 65·49 | 910 | 93·91 | 90 | 118·50 |
| 160 | 29·47 | 540 | 66·31 | 920 | 94·59 | 1300 | 119·11 |
| 170 | 30·69 | 550 | 67·13 | 930 | 95·28 | 10 | 119·72 |
| 180 | 31·88 | 560 | 67·94 | 940 | 95·96 | 20 | 120·33 |
| 190 | 33·05 | 570 | 68·74 | 950 | 96·64 | 30 | 120·94 |
| 200 | 34·21 | 580 | 69·54 | 960 | 97·32 | 40 | 121·55 |
| 210 | 35·33 | 590 | 70·34 | 970 | 97·99 | 50 | 122·15 |
| 220 | 36·44 | 600 | 71·13 | 980 | 98·66 | 60 | 122·75 |
| 230 | 37·54 | 610 | 71·92 | 990 | 99·33 | 70 | 123·35 |
| 240 | 38·62 | 620 | 72·71 | 1000 | 100·00 | 80 | 123·95 |
| 250 | 39·68 | 630 | 73·49 | 10 | 100·66 | 90 | 124·55 |
| 260 | 40·74 | 640 | 74·26 | 20 | 101·33 | 1400 | 125·14 |
| 270 | 41·78 | 650 | 75·03 | 30 | 101·99 | 10 | 125·74 |
| 280 | 42·80 | 660 | 75·80 | 40 | 102·65 | 20 | 126·33 |
| 290 | 43·81 | 670 | 76·57 | 50 | 103·30 | 30 | 126·92 |
| 300 | 44·81 | 680 | 77·33 | 60 | 103·96 | 40 | 127·51 |
| 310 | 45·80 | 690 | 78·08 | 70 | 104·61 | 50 | 128·10 |
| 320 | 46·78 | 700 | 78·84 | 80 | 105·26 | 60 | 128·69 |
| 330 | 47·75 | 710 | 79·59 | 90 | 105·91 | 70 | 129·28 |
| 340 | 48·71 | 720 | 80·33 | 1100 | 106·56 | 80 | 129·87 |
| 350 | 49·66 | 730 | 81·07 | 10 | 107·20 | 90 | 130·45 |
| 360 | 50·61 | 740 | 81·81 | 20 | 107·85 | 1500 | 131·03 |
| 370 | 51·54 | 750 | 82·55 | 30 | 108·49 | 10 | 131·61 |
| 380 | 52·46 | 760 | 83·28 | 40 | 109·13 | 20 | 132·19 |
| 390 | 53·38 | 770 | 84·01 | 50 | 109·76 | 30 | 132·77 |
| 400 | 54·29 | 780 | 84·73 | 60 | 110·40 | 40 | 133·35 |
| 410 | 55·19 | 790 | 85·46 | 70 | 111·03 | 50 | 133·93 |
| 420 | 56·08 | 800 | 86·18 | 80 | 111·67 | 60 | 134·50 |
| 430 | 56·97 | 810 | 86·89 | 90 | 112·30 | 70 | 135·08 |
| 440 | 57·85 | 820 | 87·61 | 1200 | 112·92 | 80 | 135·65 |
| 450 | 58·72 | 830 | 88·32 | 10 | 113·55 | 90 | 136·23 |
| 460 | 59·59 | 840 | 89·03 | 20 | 114·17 | 1600 | 136·80 |
| 470 | 60·45 | 850 | 89·73 | 30 | 114·80 | 10 | 137·37 |

## Table XVIII.—Two-thirds powers of numbers—*continued.*

| Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. |
|---|---|---|---|---|---|---|---|
| 1620 | 137·93 | 2080 | 162·94 | 2920 | 204·28 | 3760 | 241·80 |
| 30 | 138·50 | 2100 | 163·99 | 40 | 205·22 | 80 | 242·65 |
| 40 | 139·06 | 20 | 165·02 | 60 | 206·15 | 3800 | 243·51 |
| 50 | 139·63 | 40 | 166·05 | 80 | 207·08 | 20 | 244·36 |
| 60 | 140·19 | 60 | 167·09 | 3000 | 208·01 | 40 | 245·22 |
| 70 | 140·75 | 80 | 168·12 | 20 | 208·93 | 60 | 246·07 |
| 80 | 141·32 | 2200 | 169·15 | 40 | 209·85 | 80 | 246·91 |
| 90 | 141·88 | 20 | 170·17 | 60 | 210·76 | 3900 | 247·76 |
| 1700 | 142·44 | 40 | 171·19 | 80 | 211·68 | 20 | 248·61 |
| 10 | 143·00 | 60 | 172·20 | 3100 | 212·59 | 40 | 249·45 |
| 20 | 143·55 | 80 | 173·22 | 20 | 213·51 | 60 | 250·29 |
| 30 | 144·11 | 2300 | 174·24 | 40 | 214·42 | 80 | 251·14 |
| 40 | 144·66 | 20 | 175·24 | 60 | 215·33 | 4000 | 251·98 |
| 50 | 145·22 | 40 | 176·25 | 80 | 216·24 | 20 | 252·82 |
| 60 | 145·77 | 60 | 177·25 | 3200 | 217·15 | 40 | 253·65 |
| 70 | 146·32 | 80 | 178·26 | 20 | 218·05 | 60 | 254·49 |
| 80 | 146·87 | 2400 | 179·26 | 40 | 218·95 | 80 | 255·33 |
| 90 | 147·42 | 20 | 180·25 | 60 | 219·85 | 4100 | 256·16 |
| 1800 | 147·97 | 40 | 181·24 | 80 | 220·75 | 20 | 257·00 |
| 10 | 148·52 | 60 | 182·23 | 3300 | 221·65 | 40 | 257·83 |
| 20 | 149·06 | 80 | 183·22 | 20 | 222·54 | 60 | 258·67 |
| 30 | 149·61 | 2500 | 184·20 | 40 | 223·44 | 80 | 259·49 |
| 40 | 150·15 | 20 | 185·18 | 60 | 224·34 | 4200 | 260·31 |
| 50 | 150·70 | 40 | 186·16 | 80 | 225·22 | 20 | 261·14 |
| 60 | 151·24 | 60 | 187·14 | 3400 | 226·11 | 40 | 261·96 |
| 70 | 151·78 | 80 | 188·11 | 20 | 226·99 | 60 | 262·78 |
| 80 | 152·32 | 2600 | 189·08 | 40 | 227·88 | 80 | 263·60 |
| 90 | 152·86 | 20 | 190·05 | 60 | 228·76 | 4300 | 264·42 |
| 1900 | 153·40 | 40 | 191·02 | 80 | 229·64 | 20 | 265·24 |
| 10 | 153·94 | 60 | 191·98 | 3500 | 230·52 | 40 | 266·06 |
| 20 | 154·47 | 80 | 192·93 | 20 | 231·40 | 60 | 266·87 |
| 30 | 155·01 | 2700 | 193·89 | 40 | 232·27 | 80 | 267·69 |
| 40 | 155·54 | 20 | 194·85 | 60 | 233·14 | 4400 | 268·51 |
| 50 | 156·08 | 40 | 195·80 | 80 | 234·02 | 20 | 269·32 |
| 60 | 156·61 | 60 | 196·75 | 3600 | 234·89 | 40 | 270·13 |
| 70 | 157·14 | 80 | 197·71 | 20 | 235·76 | 60 | 270·95 |
| 80 | 157·68 | 2800 | 198·66 | 40 | 236·62 | 80 | 271·76 |
| 90 | 158·21 | 20 | 199·60 | 60 | 237·49 | 4500 | 272·56 |
| 2000 | 158·74 | 40 | 200·54 | 80 | 238·36 | 20 | 273·37 |
| 20 | 159·79 | 60 | 201·48 | 3700 | 239·22 | 40 | 274·17 |
| 40 | 160·84 | 80 | 202·42 | 20 | 240·08 | 60 | 274·98 |
| 60 | 161·89 | 2900 | 203·35 | 40 | 240·94 | 80 | 275·78 |

Table XVIII.—Two-thirds powers of numbers—*continued.*

| Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. |
|---|---|---|---|---|---|---|---|
| 4600 | 276·58 | 6150 | 335·67 | 8300 | 409·93 | 10,900 | 491·61 |
| 20 | 277·39 | 6200 | 337·49 | 50 | 411·57 | 11,000 | 494·61 |
| 40 | 278·19 | 50 | 339·30 | 8400 | 413·22 | 100 | 497·60 |
| 60 | 278·99 | 6300 | 341·11 | 50 | 414·85 | 200 | 500·58 |
| 80 | 279·78 | 50 | 342·91 | 8500 | 416·49 | 300 | 503·56 |
| 4700 | 280·58 | 6400 | 344·71 | 50 | 418·12 | 400 | 506·53 |
| 20 | 281·38 | 50 | 346·50 | 8600 | 419·75 | 500 | 509·48 |
| 40 | 282·17 | 6500 | 348·29 | 50 | 421·37 | 600 | 512·43 |
| 60 | 282·96 | 50 | 350·07 | 8700 | 423·00 | 700 | 515·38 |
| 80 | 283·76 | 6600 | 351·85 | 50 | 424·62 | 800 | 518·31 |
| 4800 | 284·55 | 50 | 353·62 | 8800 | 426·24 | 900 | 521·23 |
| 20 | 285·33 | 6700 | 355·39 | 50 | 427·85 | 12,000 | 524·15 |
| 40 | 286·11 | 50 | 357·16 | 8900 | 429·46 | 100 | 527·05 |
| 60 | 286·90 | 6800 | 358·93 | 50 | 431·06 | 200 | 529·95 |
| 80 | 287·68 | 50 | 360·68 | 9000 | 432·67 | 300 | 532·83 |
| 4900 | 288·47 | 6900 | 362·43 | 50 | 434·27 | 400 | 535·72 |
| 20 | 289·26 | 50 | 364·18 | 9100 | 435·86 | 500 | 538·60 |
| 40 | 290·05 | 7000 | 365·93 | 50 | 437·45 | 600 | 541·48 |
| 60 | 290·84 | 50 | 367·67 | 9200 | 439·04 | 700 | 544·34 |
| 80 | 291·62 | 7100 | 369·41 | 50 | 440·64 | 800 | 547·20 |
| 5000 | 292·40 | 50 | 371·13 | 9300 | 442·23 | 900 | 550·04 |
| 50 | 294·34 | 7200 | 372·86 | 50 | 443·82 | 13,000 | 552·88 |
| 5100 | 296·27 | 50 | 374·58 | 9400 | 445·40 | 100 | 555·70 |
| 50 | 298·21 | 7300 | 376·31 | 50 | 446·97 | 200 | 558·53 |
| 5200 | 300·15 | 50 | 378·02 | 9500 | 448·54 | 300 | 561·35 |
| 50 | 302·06 | 7400 | 379·74 | 50 | 450·11 | 400 | 564·16 |
| 5300 | 303·98 | 50 | 381·44 | 9600 | 451·68 | 500 | 566·96 |
| 50 | 305·89 | 7500 | 383·15 | 50 | 453·25 | 600 | 569·76 |
| 5400 | 307·80 | 50 | 384·85 | 9700 | 454·82 | 700 | 572·54 |
| 50 | 309·68 | 7600 | 386·55 | 50 | 456·39 | 800 | 575·33 |
| 5500 | 311·58 | 50 | 388·24 | 9800 | 457·95 | 900 | 578·10 |
| 50 | 313·46 | 7700 | 389·93 | 50 | 459·50 | 14,000 | 580·88 |
| 5600 | 315·34 | 50 | 391·62 | 9900 | 461·06 | 100 | 583·63 |
| 50 | 317·21 | 7800 | 393·30 | 50 | 462·61 | 200 | 586·38 |
| 5700 | 319·09 | 50 | 394·98 | 10,000 | 464·16 | 300 | 589·13 |
| 50 | 320·95 | 7900 | 396·66 | 100 | 467·25 | 400 | 591·88 |
| 5800 | 322·81 | 50 | 398·33 | 200 | 470·33 | 500 | 594·61 |
| 50 | 324·66 | 8000 | 400·00 | 300 | 473·39 | 600 | 597·34 |
| 5900 | 326·51 | 50 | 401·66 | 400 | 476·44 | 700 | 600·07 |
| 50 | 328·35 | 8100 | 403·32 | 500 | 479·49 | 800 | 602·80 |
| 6000 | 330·19 | 50 | 404·97 | 600 | 482·54 | 900 | 605·51 |
| 50 | 332·02 | 8200 | 406·63 | 700 | 485·57 | 15,000 | 608·22 |
| 6100 | 333·85 | 50 | 408·28 | 800 | 488·60 | 15,100 | 610·92 |

## Table XVIII.—Two-thirds powers of numbers—*continued.*

| Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. |
|---|---|---|---|---|---|---|---|
| 15,200 | 613·61 | 19,500 | 724·5 | 23,800 | 827·4 | 28,100 | 924·3 |
| 300 | 616·30 | 600 | 726·9 | 900 | 829·7 | 200 | 926·5 |
| 400 | 618·98 | 700 | 729·4 | 24,000 | 832·0 | 300 | 928·7 |
| 500 | 621·66 | 800 | 731·9 | 100 | 834·4 | 400 | 930·9 |
| 600 | 624·33 | 900 | 734·4 | 200 | 836·7 | 500 | 933·0 |
| 700 | 627·00 | 20,000 | 736·8 | 300 | 839·0 | 600 | 935·2 |
| 800 | 629·66 | 100 | 739·2 | 400 | 841·3 | 700 | 937·4 |
| 900 | 632·32 | 200 | 741·7 | 500 | 843·6 | 800 | 939·6 |
| 16,000 | 634·97 | 300 | 744·2 | 600 | 845·9 | 900 | 941·7 |
| 100 | 637·61 | 400 | 746·6 | 700 | 848·1 | 29,000 | 943·9 |
| 200 | 640·24 | 500 | 749·0 | 800 | 850·4 | 100 | 946·1 |
| 300 | 642·87 | 600 | 751·5 | 900 | 852·7 | 200 | 948·3 |
| 400 | 645·50 | 700 | 753·9 | 25,000 | 855·0 | 300 | 950·4 |
| 500 | 648·12 | 800 | 756·3 | 100 | 857·3 | 400 | 952·6 |
| 600 | 650·74 | 900 | 758·8 | 200 | 859·5 | 500 | 954·7 |
| 700 | 653·35 | 21,000 | 761·2 | 300 | 861·8 | 600 | 956·9 |
| 800 | 655·96 | 100 | 763·0 | 400 | 864·1 | 700 | 959·1 |
| 900 | 658·56 | 200 | 766·0 | 500 | 866 4 | 800 | 961·2 |
| 17,000 | 661·15 | 300 | 768·4 | 600 | 868·6 | 900 | 963·4 |
| 100 | 663·74 | 400 | 770·8 | 700 | 870·9 | 30,000 | 965 |
| 200 | 666·33 | 500 | 773·2 | 800 | 873·1 | 250 | 970 |
| 300 | 668·91 | 600 | 775·6 | 900 | 875·4 | 500 | 976 |
| 400 | 671·48 | 700 | 778·0 | 26,000 | 877·6 | 750 | 981 |
| 500 | 674·05 | 800 | 780·6 | 100 | 879·9 | 31,000 | 987 |
| 600 | 676·62 | 900 | 782·8 | 200 | 882·1 | 250 | 992 |
| 700 | 679·18 | 22,000 | 785·1 | 300 | 884·4 | 500 | 997 |
| 800 | 681·74 | 100 | 787·5 | 400 | 886·6 | 750 | 1002 |
| 900 | 684·29 | 200 | 789·9 | 500 | 888·9 | 32,000 | 1008 |
| 18,000 | 686·83 | 300 | 792·3 | 600 | 891·1 | 250 | 1013 |
| 100 | 689·37 | 400 | 794·6 | 700 | 893·3 | 500 | 1018 |
| 200 | 691·91 | 500 | 797·0 | 800 | 895·6 | 750 | 1023 |
| 300 | 694·44 | 600 | 799·4 | 900 | 897·8 | 33,000 | 1029 |
| 400 | 696·97 | 700 | 801·7 | 27,000 | 900·0 | 250 | 1034 |
| 500 | 699·49 | 800 | 804·1 | 100 | 902·2 | 500 | 1039 |
| 600 | 702·01 | 900 | 806·4 | 200 | 904·4 | 750 | 1045 |
| 700 | 704·52 | 23,000 | 808·8 | 300 | 906·7 | 34,000 | 1050 |
| 800 | 707·03 | 100 | 811·1 | 400 | 908·8 | 250 | 1055 |
| 900 | 709·54 | 200 | 813·4 | 500 | 911·1 | 500 | 1060 |
| 19,000 | 712·0 | 300 | 815·8 | 600 | 913·3 | 750 | 1065 |
| 100 | 714·5 | 400 | 818·1 | 700 | 915·5 | 35,000 | 1070 |
| 200 | 717·0 | 500 | 820·4 | 800 | 917·7 | 500 | 1080 |
| 300 | 719·5 | 600 | 822·8 | 900 | 919·9 | 36,000 | 1090 |
| 400 | 722·0 | 700 | 825·1 | 28,000 | 922·1 | 500 | 1100 |

**Table XVIII.—Two-thirds powers of numbers—*continued.***

| Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. | Number. | ⅔rd power. |
|---|---|---|---|---|---|---|---|
| 37,000 | 1110 | 40,500 | 1179 | 44,000 | 1246 | 50,000 | 1357 |
| 500 | 1120 | 41,000 | 1189 | 500 | 1256 | 51,000 | 1375 |
| 38,000 | 1130 | 500 | 1199 | 45,000 | 1265 | 52,000 | 1393 |
| 500 | 1140 | 42,000 | 1208 | 46,000 | 1284 | 53,000 | 1411 |
| 39,000 | 1150 | 500 | 1218 | 47,000 | 1302 | 54,000 | 1428 |
| 500 | 1160 | 43,000 | 1227 | 48,000 | 1321 | 55·000 | 1446 |
| 40,000 | 1170 | 500 | 1237 | 49,000 | 1339 | 56·000 | 1463 |

## TRIPLE VERSUS COMPOUND ENGINES.

The great economy resulting from the use of high-pressure steam is due to the fact that the increased pressure is obtained by the expenditure of an amount of heat quite insignificant compared with the additional amount of energy it renders available. The following tabular statement will make this clear at a glance :—

**Table XIX.—Steam Pressures and Efficiencies.**

| Working pressures lbs. per sq. in. | Absolute pressure lbs. per sq. in. | Heat added commences from | Evaporation effected at | Total heat of evaporation in Thermal units per pound. | Specific volume cubic feet per pound. | Total effective external work by one pound of steam in ft.-lbs. | Heat used per 1000 ft.-lbs. of work done. Thermal units. | Theoretical saving of fuel at each step per cent. |
|---|---|---|---|---|---|---|---|---|
| 30 | 45 | 100° F. | 274° F. | 1098 | 9·26 | 126,256 | 8·696 | ... |
| 75 | 90 | 100° F. | 320° F. | 1112 | 4·81 | 174,472 | 6·373 | 26·7 |
| 125 | 140 | 100° F. | 353° F. | 1122 | 3·17 | 206,994 | 5·420 | 14·9 |
| 175 | 190 | 100° F. | 377° F. | 1129 | 2·38 | 231,200 | 4·833 | 10·8 |
| 225 | 240 | 100° F. | 397° F. | 1135 | 1·912 | 255,000 | 4·425 | 8·4 |

*Note.*—The release pressure is taken to be 10 lbs. per square inch, absolute, in each case, and a back pressure of 4 lbs. per square inch, absolute, is assumed.

That is to say,—an engine using steam of 125 lbs. pressure should, other things being equal, consume nearly 15 per cent. less fuel than one using steam of 75 lbs. pressure; and one using steam of 175 lbs. pressure should effect a further saving of nearly 11 per cent.,—so that, by using 175 lbs. steam in place of 75 lbs., a saving of over 23 per cent. should be effected.

In practice, however, a two-stage compound engine, using steam of 140 lbs. pressure, shows very little economy over a simple engine using 90 lbs. steam; whilst a triple-stage engine using 140 lbs. steam shows even a greater economy than is theoretically due to the increased pressure. This is due chiefly to the diminished range of temperature in each cylinder, and the low pressure on pins, guides, and bearings.

The actual amount of fuel saved by using a triple engine, working at 160 lbs. pressure, in place of a compound engine using, say, 75 lbs. steam, is nearly 25 per cent.; and a quadruple engine with 200 lbs. saves more still, due to the higher pressure and greater number of stages of expansion.

When the average magnitudes of the stresses set up in the two types of engine are compared, the result is again greatly in favour of the triple.

Compare a three-crank triple engine with an ordinary compound, having same size of L.P. cylinder, same length of stroke, and developing the same power,—the one using steam at 150 lbs., and the other at 75 lbs.; also let the referred mean pressure be 24 lbs. in each case, and the efficiency of the expansion be the same in the two cases. Then, further, let the L.P. piston area be represented in each case by the number 14; the H.P. area of triple by 2, and the H.P. area of compound by 4, and the M.P. area of triple by 5. Then, if the equivalent, or referred, mean pressure is equally obtained from the cylinders in each case, the relative work done will be as follows:—

| **Triple Engine.** | **Compound Engine.** |
|---|---|
| H.P. cylinder, $2 \times \left(\frac{24}{3} \times \frac{14}{2}\right)$ or 112. | H.P. cylinder, $4 \times \left(\frac{24}{2} \times \frac{14}{4}\right)$ or 168. |
| M.P. ,, $5 \times \left(\frac{24}{3} \times \frac{14}{5}\right)$ or 112. | L.P. ,, $14 \times \frac{24}{2}$ or 168. |
| L.P. ,, $14 \times \frac{24}{3}$ or 112. | |

That is,—the average stress on the rods, columns, guides, &c., is 50 per cent. more with the compound than with the triple engine. The triple engine is also better balanced than the compound, works with less vibration, gives a more equable turning moment, and consequently a higher efficiency of propeller.

It is not surprising, therefore, that the triple compares so favourably with the compound as regards wear and tear.

## RATIOS OF CYLINDERS.

**Two-Stage** or **Compound Engines.**—Engines of this type are still commonly fitted in paddle vessels, and in small screw steamers; as also for many auxiliary purposes, such as for driving independent air and circulating pumps, feed pumps, dynamos, &c.

When fitted in **Paddle vessels**, the boiler pressure is usually from 70 to 100 lbs. for oscillating engines, and from 90 to 120 lbs. for other types,—the latter pressure mostly with diagonal engines.

But considerations of weight and bulk preclude the adoption of high ratios, and experience has shown that a ratio of about 1 : 3·25 for pressures up to 110 lbs., and for 140 lbs. 1 : 3·8 is quite satisfactory. The extra quantity of coal to be carried is of little consequence where the voyage is only three or four hours' duration.

The ratios of cylinders for **auxiliary engines** are also determined more from considerations of weight and bulk than of economy, and with little regard to boiler pressures,—with the result that they are rarely made so great as in the case of the propelling engines, even when the auxiliary engine must run hour for hour with the main engines; in practice this ratio is generally about 1 : 2·5 to 1 : 3 for cylinders side by side, and 1 : 3·5 to 1 : 4 for tandem engines. In Naval practice, when the boiler pressure used was 300 lbs., the cylinders of compound auxiliary engines were only made 4 to 1.

**Three-stage** or **Triple engines,** and **Four-Stage** or **Quadruple engines.**—In the case of main or propelling engines for merchant or mail steamers, space and weight are, within ordinary limits, of very little consequence, whilst coal consumption is of the utmost importance; L.P. cylinders are therefore usually large enough to expand the steam to the full economical limit,—of say, 6 lbs. absolute.

The ratios of cylinders depend largely on the rate of expansion at full speed. The following holds good in practice:—

$$\text{Rate of expansion} = \frac{\text{absolute pressure}}{Q}.$$

For Naval engines, $Q = 21$.
In express steamers short voyage, $Q = 19$.
,, ,, long ,, $Q = 18$.
Passenger cargo steamers, $Q = 17$.
Tramp cargo steamers, $Q = 16$.

The following Table shows the ratios of cylinders necessary to effect this, with a cut-off in the H.P. of ·6 of the stroke:—

### Table XX.—Ratios of Cylinders in multiple Stage Reciprocators.

| Absolute pressure. | 125 | 135 | 145 | 155 | 165 | 175 | 185 | 195 | 205 | 215 | 225 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Ratio $\frac{\text{M.P.}}{\text{H.P.}}$ | 2·04 | 2·11 | 2·19 | 2·26 | 2·33 | 2·40 | 2·47 | 2·54 | 2·61 | 2·68 | 2·75 |
| Ratio $\frac{\text{L.P.}}{\text{H.P.}}$ | 4·4 | 4·75 | 5·1 | 5·43 | 5·8 | 6·13 | 6·5 | 6·8 | 7·18 | 7·53 | 7·9 |
| Working pressure (above atmosphere) | 110 | 120 | 130 | 140 | 150 | 160 | 170 | 180 | 190 | 200 | 210 |

The values given in the above table for the ratio $\frac{\text{L.P.}}{\text{H.P.}}$ are such that the nominal rate of expansion is, in each case, $\frac{\text{absolute pressure}}{17}$; and those for ratio $\frac{\text{M.P.}}{\text{L.P.}}$ are calculated by the formula,—

**Rule 31.** $$\text{M.P. area} = \frac{\text{L.P. area}}{1{\cdot}1 \times \sqrt{\text{Ratio of H.P. to L.P.}}}$$

When the working pressure exceeds 195 lbs. absolute for economy in consumption, the steam should be expanded in four stages, the engine should be a Quadruple.

For Quadruple engines, working at 200 lbs. to 215 lbs. pressure, the relative areas of cylinders may be about 1 : 1·8 : 3·6 : 7·9;

$$\text{or } \frac{\text{1st M.P.}}{\text{H.P.}} = 1{\cdot}8,$$

$$\frac{\text{2nd M.P.}}{\text{1st M.P.}} = 2,$$

$$\text{and } \frac{\text{L.P.}}{\text{2nd M.P.}} = 2{\cdot}2.$$

It is possible to arrange the cylinders of large engines in many different ways, but the designer must always keep in view equality of stresses, ranges of temperature in the various cylinders, and also the best disposition of the reciprocating weights for reducing vibration.

Figs. 8, 8A, 8B, and 8C show some of these arrangements.

If possible, three cranks should be employed in preference to two, and the tandem arrangement only be resorted to in cases where there would otherwise be a low-pressure cylinder of very large size,—say over 100 inches diameter, and where four cranks are inadmissible.

**Naval Engines.**—In war-vessels economy at full power is only a secondary consideration, whilst weight of machinery and space occupied by it are of the first importance, and the ratio of L.P. piston to H.P. is therefore seldom so great as in the merchant service. While the boiler pressure used in the Navy stood at 155 lbs., the piston areas were usually made of the proportions 1 : 2 to 2·25 : 4·7 to 5. With a working pressure of 250 lbs. at the engines the proportions used have been about 1 : 2·65 : 7. In passenger steamers and war-vessels with relatively high revolution speeds, the necessity of balancing the inertia forces to reduce vibration to a minimum has led to the use of four-crank triple and four-crank quadruple engines.

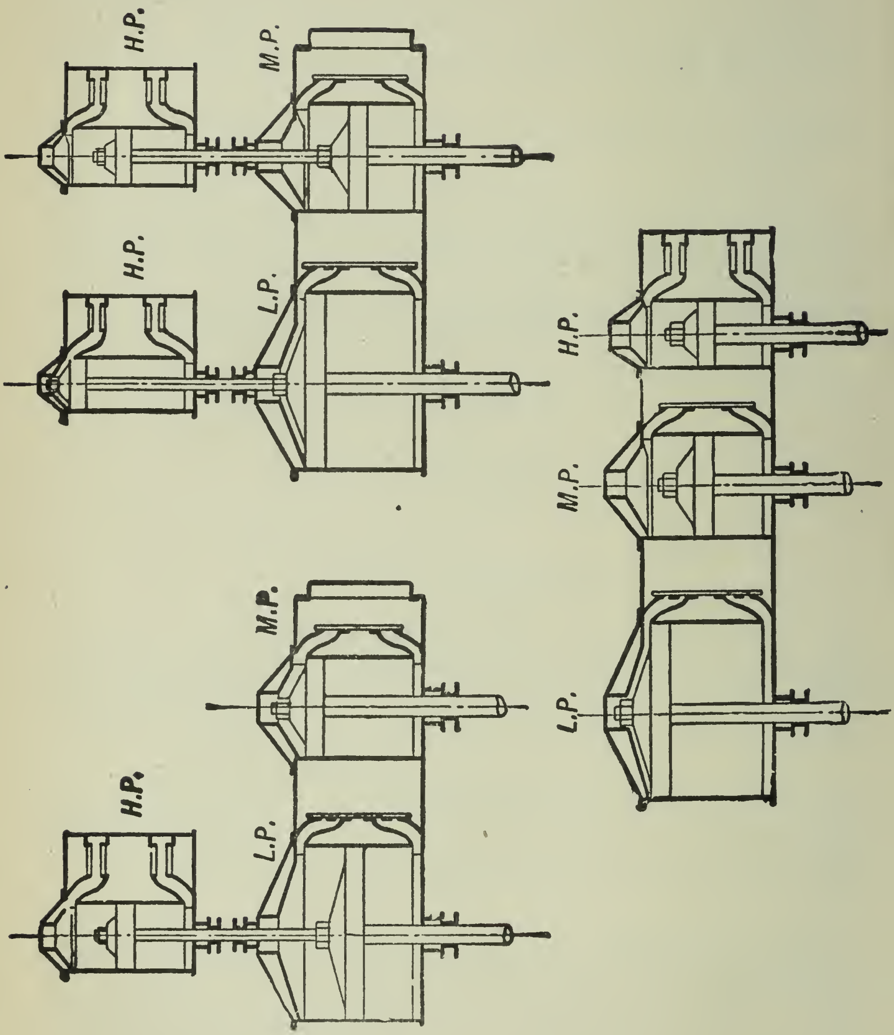


FIG. 8. Triple Expansion Engines.

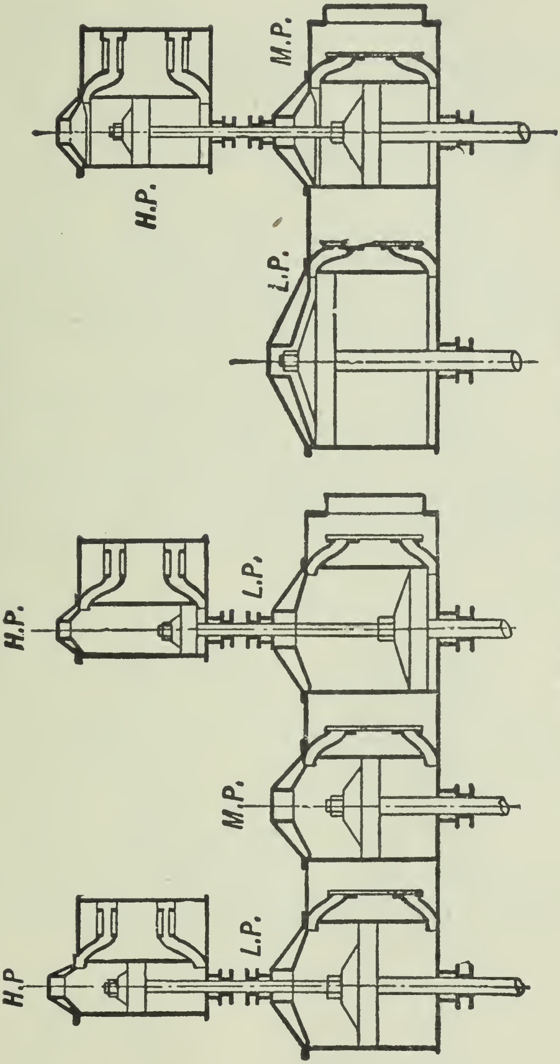


FIG. 8A. Triple Expansion Engines.

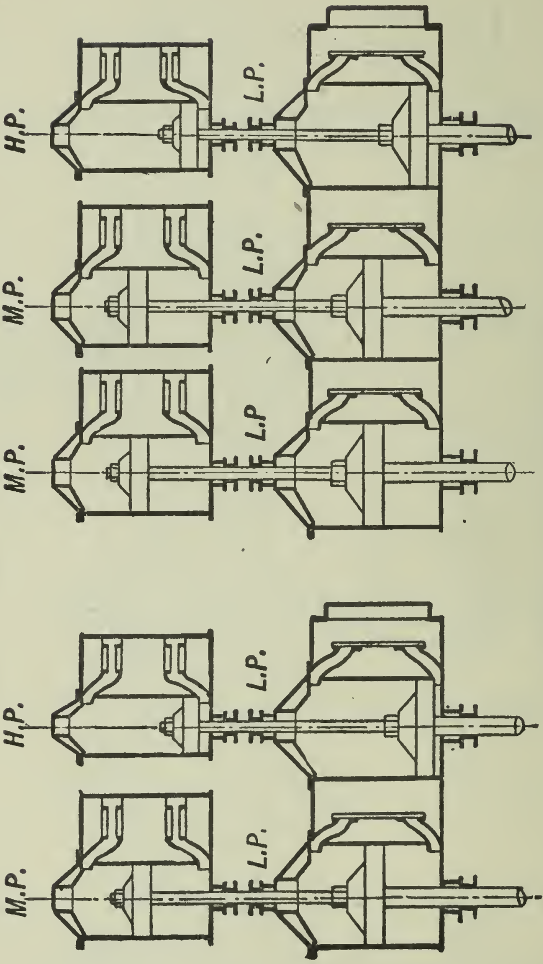


FIG. 38. Triple Expansion Engines.

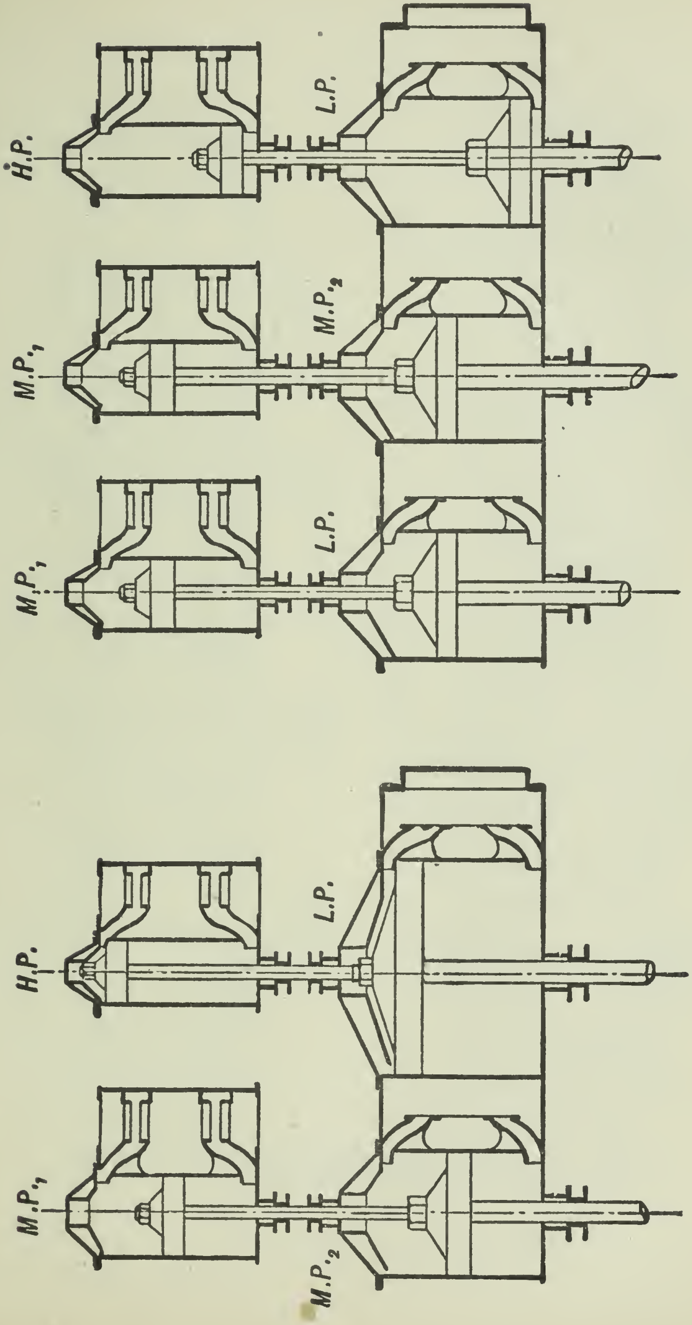


FIG. 8C. Quadruple Expansion Engines.

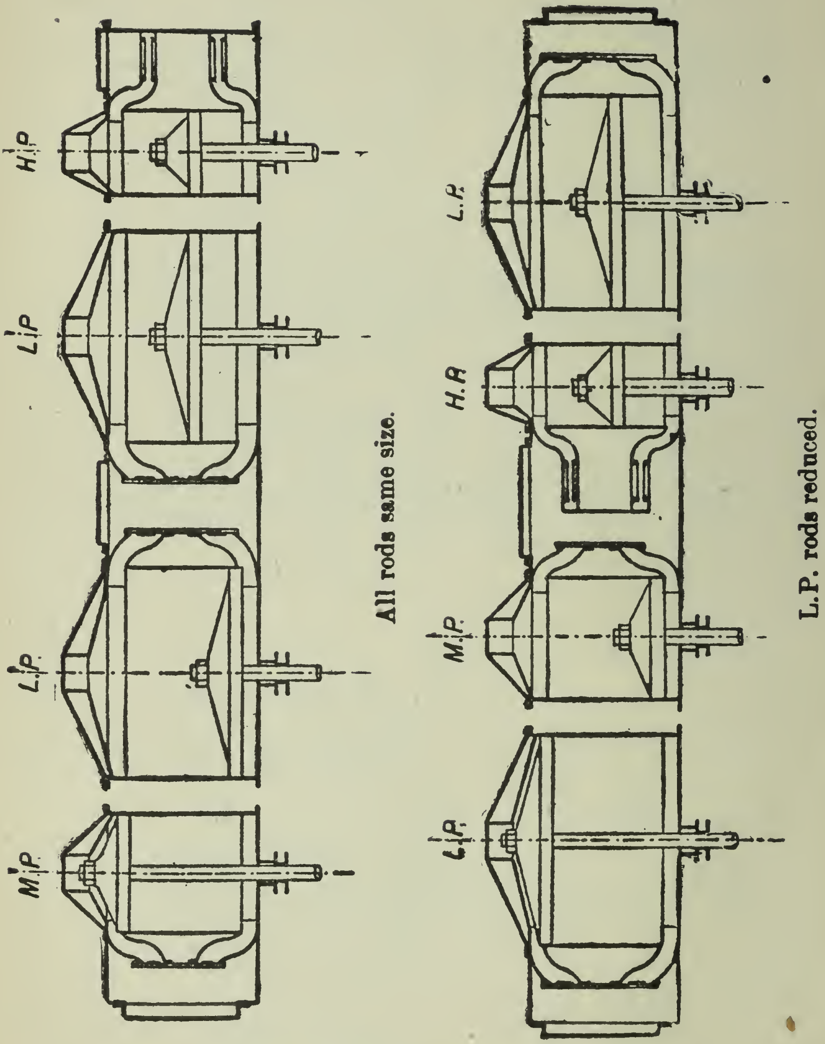
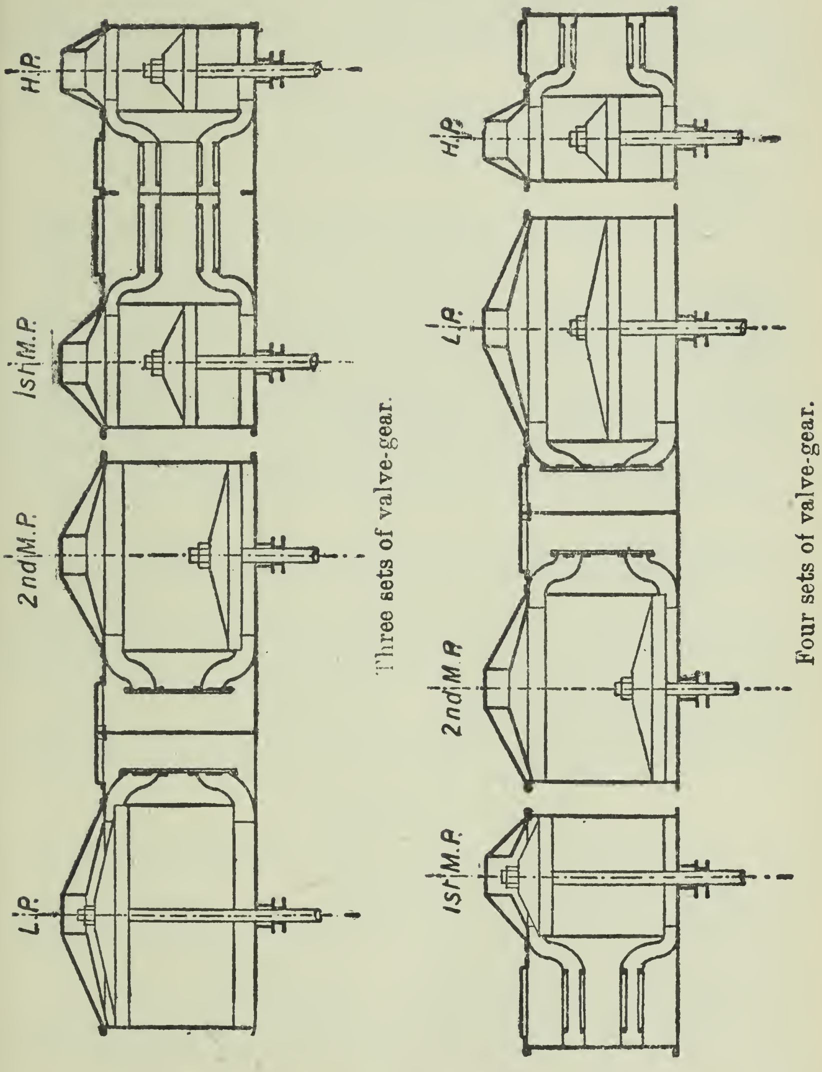


FIG. 8D.—Four-crank Triple Expansion Engines.



Three sets of valve-gear.

Four sets of valve-gear.

FIG. 8E.—Four-crank Quadruple Expansion Engines.

## EQUIVALENT MEAN PRESSURES, ETC.

In estimating the mean pressure in any multiple-stage engine, it is usual *to refer it to the L.P. cylinder*, *i.e.* to calculate the pressure that would be required if the work were all done in the L.P. cylinder only.

The value of this equivalent—or, as it is called, *referred*—mean pressure for a triple engine is therefore,—

**Rule 32.**

$$Rp_m = \text{L.P. mean press.} + \frac{\text{M.P. mean press.}}{\text{Ratio of L.P. to M.P.}} + \frac{\text{H.P. mean press.}}{\text{Ratio of L.P. to H.P.}}$$

Records of the $Rp_m$ obtained with each engine should be kept,—together with notes of the various conditions under which the trials were made,—such as pressure of steam at H.P. valve casing, degree of opening of throttle or regulator valve, amount gear is "linked-up," vacuum in condenser, &c.,—as they supply a ready means of checking calculations for sizes of cylinders, cuts off, &c., of proposed engines of the same type, and of determining the exact values of the factors, approximate values of which are given in Table XXIV. page 73.

### Expansion of Steam: to determine Mean Pressures.

The mean pressure obtained, and consequently the work done, in any steam cylinder, or series of steam cylinders, depends mainly on the absolute pressure and the condition of the steam at admission, also the rate of expansion at release.

The general formula for pressure and volume of steam expanding under conditions obtaining in practice is

$$pv^{\epsilon} = \text{constant.}$$

When the cylinders are jacketed so as to render heat to the steam rather than abstract it during expansion,—

(i.) Steam as supplied from the ordinary boiler without superheater or other means of drying, $\epsilon = 1{\cdot}00$.

(ii.) When the steam is supplied to such cylinders "dry" and there is no condensation during expansion, $\epsilon = 1{\cdot}0625$ or $^{17}/_{16}$ths.

(iii.) When the steam to these cylinders is superheated and the temperature is kept constant during expansion (*isothermic*), then $\epsilon = 1{\cdot}00$.

When the cylinders are not jacketed but well clothed, as in general marine practice,—

(iv.) Steam supplied simply as produced saturated $\epsilon = 1{\cdot}111$, ($^{10}/_{9}$ths).

(v.) If superheated steam supplied $\epsilon = 1{\cdot}180$.

## Mean Pressure of Steam Expanding Isothermally, as in a Steam Cylinder when Jacketed.

Let V be the volume, P the pressure, and T the temperature of steam as supplied.

,, $v$ be the volume, $p$ the pressure, $t$ the temperature at the end of expansion.

,, $v_0$ be the volume, $p_0$ the pressure, and $t_0$ the temperature of back pressure.

R is the ratio of expansion $=\frac{P}{p}$; and PV = constant.

A. (1) Mean pressure due to a pressure P, and a rate of expansion R.

**Rule 33.** $P_m = \frac{1+\log_\epsilon R}{R} \times P - p_0$ ($p_0 = o$, *vide* Table XXI.)

If the back pressure in a modern condenser be taken as 1 lb., then for a full expansion, as in the case of a turbine, R = P.

(2) $P_m = \frac{1+\log_\epsilon R}{R} \times R - 1$; or $P_m = \log_\epsilon R$ or $\log_\epsilon P$.

(3) The work done by a pound of steam during admission and expansion from pressure P to 1 lb. will be found as follows:—

Maximum work of 1 lb. of steam of pressure $P = \log_\epsilon P \times 144 \times v$; in this case $v = 332$. Then

$$\text{maximum work} = 47{,}808 \log_\epsilon P.$$

(4) If steam expands to a pressure $p$ and then drops to $p_0$, which is the usual condition of working in a reciprocating engine,

$$\text{Mean pressure} = \frac{1+\log_\epsilon R}{R} \times P - \frac{1+\log_\epsilon p}{r} \times p + (p - p_0),$$

assuming the final pressure of the curve of expansion if continued to be 1 lb.,

$$\frac{1+\log_\epsilon R}{R} \times P = (1+\log_\epsilon P) \text{ or } (1+\log_\epsilon R)$$

$\frac{1+\log r}{r} \times p$ will likewise be equal to $(1+\log_\epsilon p)$.

$$\text{Then } P_m = (1+\log_\epsilon P) - (1+\log_\epsilon p) + (v - p_0)$$

$$= \log_\epsilon P - \log_\epsilon p + p - p_0,$$

since $p_0 = 1$. Here again $P_m = \log_\epsilon P - \log_\epsilon p + p$.

The maximum work possible from 1 lb. of steam under these conditions,

$$=(\text{Log}_\epsilon \text{P} - \log_\epsilon p + p)\ 47{,}808 \text{ foot-lbs.}$$

(5) If steam expands to a pressure $p$, and exhausts into a receiver in which the pressure is $p_1$, then

$$\text{P}_m = \log_\epsilon \text{P} - \log_\epsilon p.$$

The maximum work done externally under these circumstances by 1 lb. of steam of a pressure $\text{P} = (\log_\epsilon \text{P} - \log_\epsilon p)$ 47,808 foot-lbs.

B. If the steam is dry and expands adiabatically then

$$pv^{17/16} = \text{constant (69,000 foot-lbs. Rankine).}$$

**Rule 33a.** Then $\text{P}_m = \dfrac{17 - 16\text{R}^{-\frac{1}{16}}}{\text{R}} \times \text{P}$ . . . . (1)

If the steam expands to pressure $p$ and drops to the pressure $p_0$,

$$\text{P}_m = \left(\frac{17 - 16\text{R}^{-\frac{1}{16}}}{\text{R}}\right) \times \text{P} - \left(\frac{17 - 16r^{-\frac{1}{16}}}{r}\right) p + (p - p_0) \quad . \quad (2)$$

If $p_0 = 1$, and steam expands to $p$, then $\text{R} = \text{P} \div 1$ and $r = p \div 1$.

$$\text{Then P}_m = (17 - 16\text{R}^{-\frac{1}{16}}) - (17 - 16r^{-\frac{1}{16}}) + (p - 1)$$

$$= 16\,(r^{-\frac{1}{16}} - \text{R}^{-\frac{1}{16}}) + (p - 1) \quad . \quad . \quad . \quad . \quad (3)$$

When steam expands adiabatically from pressure P to 1 lb., and it condenses at pressure $p$, then

Maximum work done by 1 lb. of steam externally,

$$= 144v\left\{(17 - 16\text{P}^{-\frac{1}{16}}) - 1\right\}$$

$$= 2304v\,(1 - \text{P}^{-\frac{1}{16}})$$

and since $v = 333$ cubic feet.

Maximum work obtainable from 1 lb. steam,

$$= 767{,}222\,(1 - \text{P}^{-\frac{1}{16}}) \text{ foot-lbs.} \quad . \quad . \quad . \quad (4)$$

Maximum work obtainable in B thermal units,

$$= (\text{T}_1 - \text{T}_2) \times \left(1 - \frac{\text{L}_1}{\text{T}_1}\right) - \text{T}_2 \log_\epsilon \frac{\text{T}_1}{\text{T}_2} \quad . \quad . \quad (5)$$

This (Sir J. Ewing, p. 100) $\text{T}_1$ is the absolute temperature, and $\text{L}_1$ the latent heat of steam at P. $\text{T}_2$ the absolute temperature to which steam expands and condenses. Table VII. is calculated by (5).

## Table XXI.—Steam used Expansively.

| R Rate of Expansion. | 1÷R Cut off. | $\frac{1+\text{hyp. log } R}{R}$ $\epsilon=1{\cdot}00$ | $\frac{17-16R^{-\frac{1}{16}}}{R}$ $\epsilon=1{\cdot}0625$ | R | $\frac{1}{R}$ | $\frac{1+\text{hyp. log } R}{R}$ $\epsilon=1{\cdot}0$ | $\frac{17-16R^{-\frac{1}{16}}}{R}$ $\epsilon={}^{17}/_{16}$ | R | $\frac{1}{R}$ | $\frac{1+\text{hyp. log } R}{R}$ $\epsilon=1{\cdot}0$ | $\frac{17-16R^{-\frac{1}{16}}}{R}$ $\epsilon={}^{17}/_{16}$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 50 | ·0200 | ·09825 | ·08942 | 20 | ·050 | ·1998 | ·186 | 4·00 | ·250 | ·5965 | ·582 |
| 48 | ·0209 | ·10150 | ·09248 | 18 | ·055 | ·2161 | ·200 | 3·63 | ·275 | ·6308 | ·616 |
| 46 | ·0218 | ·10498 | ·09580 | 16 | ·062 | ·2358 | ·220 | 3·33 | ·300 | ·6615 | ·648 |
| 44 | ·0227 | ·10875 | ·09920 | 15 | ·066 | ·2472 | ·230 | 3·00 | ·333 | ·6993 | ·670 |
| 42 | ·0238 | ·11282 | ·10317 | 14 | ·071 | ·2599 | ·242 | 2·86 | ·350 | ·7171 | ·707 |
| 40 | ·0250 | ·11724 | ·10735 | 13·33 | ·075 | ·2690 | ·254 | 2·66 | ·375 | ·7440 | ·733 |
| 38 | ·0263 | ·12206 | ·11195 | 13 | ·077 | ·2742 | ·258 | 2·50 | ·400 | ·7664 | ·756 |
| 36 | ·0278 | ·12734 | ·11700 | 12 | ·083 | ·2904 | ·271 | 2·22 | ·450 | ·8095 | ·800 |
| 34 | ·0294 | ·13314 | ·12380 | 11 | ·091 | ·3089 | ·292 | 2·00 | ·500 | ·8465 | ·840 |
| 32 | ·0312 | ·13957 | ·12918 | 10 | ·100 | ·3303 | ·314 | 1·82 | ·550 | ·8786 | ·874 |
| 30 | ·0333 | ·14673 | ·13600 | 9 | ·111 | ·3552 | ·340 | 1·66 | ·600 | ·9066 | ·900 |
| 28 | ·0357 | ·15474 | ·14351 | 8 | ·125 | ·3849 | ·370 | 1·60 | 625 | ·9187 | ·913 |
| 26 | ·0385 | ·16380 | ·15154 | 7 | ·143 | ·4210 | ·408 | 1·54 | ·650 | ·9292 | ·926 |
| 25 | ·0400 | ·16877 | ·15611 | 6·66 | ·150 | ·4347 | ·417 | 1·48 | ·675 | ·9405 | ·940 |
| 24 | ·0417 | ·17411 | ·16113 | 6·00 | ·166 | ·4653 | ·450 | 1·43 | ·700 | ·950 | ·945 |
| 23 | ·0435 | ·17982 | ·16661 | 5·71 | ·175 | ·4807 | ·466 | 1·33 | ·750 | ·965 | ·960 |
| 22 | ·0455 | ·18598 | ·17275 | 5·00 | ·200 | ·5218 | ·506 | 1·25 | ·800 | ·978 | ·976 |
| 21 | ·0476 | ·19260 | ·17955 | 4·44 | ·225 | ·5608 | ·540 | 1·18 | ·850 | ·989 | ·986 |

## Table XXII.—Factors for Finding the Theoretical Mean Pressures when Clearance Volumes are taken into Account.

| R Rate of Expansion. | $\frac{1}{R}$ Cut off. | Clearance Volume per cent. of Cylinder. | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | 4·0 | 6·0 | 8·0 | 10·0 | 12·0 | 14·0 | 16·0 |
| 20·0 | 0·05 | 0·28 | 0·30 | 0·32 | 0·35 | 0·37 | 0·39 | 0·41 |
| 16·67 | 0·06 | 0·30 | 0·32 | 0·34 | 0·37 | 0·39 | 0·41 | 0·42 |
| 14·29 | 0·07 | 0·32 | 0·34 | 0·36 | 0·39 | 0·41 | 0·43 | 0·44 |
| 12·50 | 0·08 | 0·34 | 0·36 | 0·38 | 0·40 | 0·42 | 0·44 | 0·46 |
| 11·11 | 0·09 | 0·36 | 0·38 | 0·40 | 0·42 | 0·44 | 0·46 | 0·48 |
| 10·00 | 0·10 | 0·38 | 0·40 | 0·42 | 0·44 | 0·46 | 0·47 | 0·50 |
| 8·33 | 0·12 | 0·42 | 0·44 | 0·45 | 0·47 | 0·49 | 0·50 | 0·52 |
| 7·14 | 0·14 | 0·45 | 0·47 | 0·49 | 0·50 | 0·52 | 0·53 | 0·54 |
| 6·25 | 0·16 | 0·48 | 0·50 | 0·52 | 0·53 | 0·55 | 0·56 | 0·57 |
| 5·56 | 0·18 | 0·51 | 0·53 | 0·55 | 0·56 | 0·57 | 0·59 | 0·60 |
| 5·00 | 0·20 | 0·56 | 0·57 | 0·58 | 0·59 | 0·60 | 0·61 | 0·62 |
| 4·00 | 0·25 | 0·62 | 0·63 | 0·64 | 0·65 | 0·66 | 0·67 | 0·68 |
| 3·33 | 0·30 | 0·68 | 0·69 | 0·70 | 0·70 | 0·71 | 0·72 | 0·73 |
| 2·86 | 0·35 | 0·73 | 0·74 | 0·75 | 0·75 | 0·76 | 0·76 | 0·77 |
| 2·50 | 0·40 | 0·77 | 0·78 | 0·79 | 0·80 | 0·80 | 0·80 | 0·81 |
| 2·22 | 0·45 | 0·81 | 0·82 | 0·83 | 0·84 | 0·84 | 0·84 | 0·85 |
| 2·00 | 0·50 | 0·85 | 0·86 | 0·86 | 0·87 | 0·87 | 0·87 | 0·88 |
| 1·75 | 0·55 | 0·89 | 0·89 | 0·89 | 0·90 | 0·90 | 0·90 | 0·90 |
| 1·67 | 0·60 | 0·91 | 0·91 | 0·91 | 0·92 | 0·92 | 0·92 | 0·92 |

The above table is for saturated steam expanding with $pv = C$.

If it is required to ascertain the theoretical mean pressure on a cylinder having 10 per cent. clearance with steam of 120 lbs. absolute pressure cut off at 0·12 of the stroke, then

Mean pressure = 120 × 0·47 or 56·4 lbs.

If exhausting to a condenser, with a vacuum of 28 ins., then

Mean pressure = 56·4 − 1 or 55·4 lbs.

### Table XXIII.—Mean Pressure of Dry Steam Expanded Adiabatically ($\epsilon = {}^{17}/_{16}$).

| Rate of expansion. | Initial Pressures (absolute). | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 25 lbs. | 50 lbs. | 75 lbs. | 100 lbs. | 125 lbs. | 150 lbs. | 175 lbs. | 200 lbs. | 225 lbs. | 250 lbs. |
| 20 | 4·65 | 9·30 | 13·95 | 18·60 | 23·25 | 27·90 | 32·55 | 37·20 | 41·85 | 46·50 |
| 19 | 4·83 | 9·66 | 14·49 | 19·32 | 24·15 | 28·98 | 33·81 | 38·64 | 43·47 | 48·30 |
| 18 | 5·02 | 10·05 | 15·06 | 20·08 | 25·09 | 30·12 | 35·15 | 40·28 | 45·27 | 50·28 |
| 17 | 5·24 | 10·48 | 15·72 | 20·96 | 26·20 | 31·44 | 36·68 | 41·92 | 47·16 | 52·40 |
| 16 | 5·50 | 11·00 | 16·50 | 22·00 | 27·50 | 33·00 | 38·50 | 44·00 | 49·50 | 55·00 |
| 15 | 5·75 | 11·50 | 17·25 | 23·00 | 28·75 | 34·50 | 40·25 | 46·00 | 51·75 | 57·50 |
| 14 | 6·05 | 12·10 | 18·15 | 24·20 | 30·25 | 36·30 | 42·35 | 48·40 | 54·45 | 60·50 |
| 13 | 6·45 | 12·90 | 19·35 | 25·80 | 32·25 | 38·70 | 45·15 | 51·60 | 58·05 | 64·50 |
| 12 | 6·78 | 13·56 | 20·34 | 27·12 | 33·90 | 40·68 | 47·46 | 54·24 | 61·02 | 67·80 |
| 11 | 7·30 | 14·60 | 21·90 | 29·20 | 36·50 | 43·80 | 51·10 | 58·40 | 65·70 | 73·00 |
| 10 | 7·85 | 15·70 | 23·55 | 31·40 | 39·25 | 47·10 | 54·95 | 62·80 | 70·65 | 78·50 |
| 9 | 8·50 | 17·00 | 25·50 | 34·00 | 42·50 | 51·00 | 59·50 | 68·00 | 76·50 | 85·00 |
| 8 | 9·25 | 18·50 | 27·75 | 37·00 | 46·75 | 55·50 | 64·75 | 74·00 | 83·25 | 92·50 |
| 7 | 10·20 | 20·40 | 30·60 | 40·80 | 51·00 | 61·20 | 71·40 | 81·60 | 91·80 | 102·00 |
| 6 | 11·25 | 22·50 | 33·75 | 45·00 | 56·25 | 67·50 | 78·75 | 90·00 | 101·25 | 112·50 |
| 5 | 12·65 | 25·30 | 37·95 | 50·60 | 63·25 | 75·90 | 88·55 | 101·20 | 113·85 | 126·50 |
| 4·5 | 13·55 | 27·10 | 40·65 | 54·20 | 67·75 | 81·30 | 94·85 | 108·40 | 121·95 | 135·50 |
| 4·0 | 14·55 | 29·10 | 43·65 | 58·20 | 72·75 | 87·30 | 101·85 | 116·40 | 130·95 | 145·50 |
| 3·5 | 15·55 | 31·10 | 46·65 | 62·20 | 77·75 | 93·30 | 108·85 | 124·04 | 139·95 | 155·50 |
| 3·0 | 16·75 | 33·50 | 50·25 | 67·00 | 83·75 | 100·50 | 117·25 | 134·00 | 150·75 | 167·50 |
| 2·5 | 18·50 | 37·00 | 55·50 | 74·00 | 92·50 | 111·00 | 129·50 | 148·00 | 166·50 | 185·00 |
| 2·0 | 21·00 | 42·00 | 63·00 | 84·00 | 105·00 | 126·00 | 147·00 | 168·00 | 189·00 | 210·00 |
| 1·82 | 21·85 | 43·70 | 65·55 | 87·40 | 109·25 | 131·10 | 152·95 | 174·80 | 196·65 | 218·50 |
| 1·66 | 22·50 | 45·00 | 67·50 | 90·00 | 112·50 | 135·00 | 157·50 | 180·00 | 202·50 | 225·00 |
| 1·60 | 22·83 | 45·76 | 68·49 | 91·32 | 114·15 | 136·98 | 159·81 | 181·04 | 205·47 | 228·30 |
| 1·54 | 23·15 | 46·30 | 69·45 | 92·60 | 115·65 | 138·90 | 162·05 | 185·20 | 208·35 | 231·50 |
| 1·48 | 23·50 | 47·00 | 70·50 | 94·00 | 117·50 | 141·00 | 164·50 | 188·00 | 211·50 | 235·00 |

Column 4, Table XXI., gives the multipliers for *dry saturated* steam, calculated by Rankine's formula $\frac{p_m}{p_1}=\frac{17-16r^{-\frac{1}{16}}}{r}$, which is based on the assumption that $pv^{\frac{17}{16}}$=constant, and this expansion is called *adiabatic* as it takes place without any added heat as in the ordinary marine engine cylinder. At very high rates of expansion only is there very great difference between column 3 and column 4.

Absolute initial pressure should be taken at 3 to 5 lbs. below the absolute boiler pressure, and back pressure at not less than 3 lbs.; also, number of expansions is to be understood as the *nominal* number, or $\frac{\text{L.P. capacity}}{\text{H.P. capacity at cut-off}}$, without regard to clearances, or to the occurrence of release in final cylinder before end of stroke.

Then, to determine the effective $p_m$, when the boiler pressure is fixed, and the best number of expansions agreed upon,—multiply the $p_1$ by the quantity found in column 3 of Table XXI. opposite the proper number of expansions, and subtract back pressure from the product.

As values of effective mean pressures, found by aid of Table XXI., are determined without taking into consideration such disturbing elements as clearance, compression, radiation, friction of ports and passages, receiver "drop," initial condensation and subsequent re-evaporation, &c., it is necessary to allow for these in some way; but, the calculations for their values being laborious, and involving so many assumptions as to be of doubtful accuracy when finished, it is best to allow for them by multiplying by a single factor, the value of which is derived from experience.

Then $p_m \times$ factor $= \text{R}p_m$, or referred mean pressure—(*q.v.* page 73).

Table XXIV. gives the average value of this factor for the various types of engine, and under the various conditions named:—

## Table XXIV.—Ratios of Mean Pressures as in Practice to Theoretical.

| Description of Engine. | | Jacketed. | Unjacketed. |
|---|---|---|---|
| (1) Expansion taking place all in one cylinder; ports of average size; and ordinary slide valve driven by ordinary eccentric;—as in various auxiliary engines, . . | . | . | ·76 to ·80<br>MEAN ·78 |
| (2) Expansion taking place all in one cylinder; ports of average size; and ordinary slide valves driven by ordinary eccentric gear;—as in low-pressure paddle engines, . | . | . | ·68 to ·75<br>MEAN ·71 |
| (3) Expansion taking place in two cylinders with receiver between; ports of average size; ordinary slide valves (H.P. single-ported, and L.P. double-ported); and ordinary eccentric gear;—as in paddle engines, small screw engines, and some auxiliary engines, . . . . | Screw | ·67 to ·73<br>MEAN ·70 | ·58 to ·68<br>MEAN ·63 |
| | Paddle | . | ·55 to ·65<br>MEAN ·60 |
| | Auxly. | . | ·45 to ·50<br>MEAN ·47 |
| (4) Expansion taking place in two cylinders, placed in same line, and exhausting direct from one to the other; ports, slide valves, and gear as in (3);—as occasionally used, . . . . . | . | . | ·71 to ·75<br>MEAN ·72 |
| (5) Expansion taking place in three cylinders, placed side by side, with receivers between them; ports of average size; ordinary slide valves (say piston valve for H.P. and double-ported flat slides for M.P. and L.P.); and ordinary eccentric gear;—as used in most merchant steamers, . . . | . | ·64 to ·68<br>MEAN ·66 | ·60 to ·66<br>MEAN ·63 |
| (6) Engines same as (5); but ratio of H.P. to L.P. not exceeding 1:5; and running at 90 to 140 revolutions;—as in ironclads and cruisers, . . . . . . | . | ·55 to ·65<br>MEAN ·60 | . |
| (7) Engines same as (5); but ratio of H.P. to L.P. not exceeding 1:5; and running at 200 to 350 revolutions;—as in torpedo gunboats, &c., . . . . . . . | . | . | ·60 to ·67<br>MEAN ·63 |

In the case of any multiple-stage engine, where the power is divided equally amongst the cylinders, whose number is N, and where the

Table XXV.—Results of Trials of Marine Engines (Twin Screw).

| Name. H.M.S. | Nature of Trials. | Cylinders. | | | | | Engine Room. | | | | | | | | | One Engine. | | | | | Nominal Rate of Expansion. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diameters. | | | | Stroke. Inches. | Steam. Lbs. | Vacuum. Inches. | Revolts. | Cut-off. H.P. | Mean Pressures. | | | | | Indicated Horse Power. | | | | | |
| | | H.P. | M.P. | L.P. | L.P. | | | | | | H.P. | M.P. | L.P. | L.P. | Ref. to L.P. Cl. | H.P. | M.P. | L.P. | L.P. | Total. | |
| Good Hope, | F.S. | 43½ | 71 | 81½ | 81½ | 48 | 278 | 27 | 126 | 0·73 | 105 | 44·6 | 16·9 | 17·9 | 49 3 | 4,764 | 5,399 | 2,698 | 2,849 | 15,710 | 9·6 |
| | R.S. | | | | | | 268 | 26·5 | 110·7 | 0·63 | 95·8 | 37·9 | 11·9 | 13·4 | 40·7 | 3,819 | 4,029 | 1,606 | 1,864 | 11,378 | .. |
| Formidable, | F.S. | 31½ | 51⅓ | 84 | .. | 51 | 245 | 25·5 | 110·2 | 0·70 | 115 | 44 | 17·9 | .. | 50·5 | 2,543 | 2,601 | 2,815 | .. | 7,959 | 10·1 |
| | R.S. | | | | | | 240 | 26·5 | 101 | .. | 95·2 | 34·4 | 14·3 | .. | 40·7 | 1,934 | 1,868 | 2,066 | .. | 5,868 | .. |
| Arrogant, | F.S. | 26 | 42 | 68 | .. | 39 | 245 | 26 | 143 | 0·70 | 112·5 | 47·1 | 20·5 | .. | 52·8 | 1,685 | 1,841 | 2,100 | .. | 5,406 | 9·8 |
| | R.S. | | | | | | 235 | 27 | 128 | 0·51 | 86 5 | 35·3 | 15·8 | .. | 41·2 | 1,159 | 1,233 | 1,449 | .. | 3,841 | 13·5 |
| Apollo, | F.D. | 33⅓ | 49 | 74 | .. | 39 | 151 | 26·1 | 137·3 | 0·85 | 51·9 | 32·7 | 13·9 | .. | 38·8 | 1,236 | 1,669 | 1,621 | .. | 4,527 | 5·7 |
| | N.D. | | | | | | 150 | 28 | 130 | .. | 49·4 | 27·3 | 12·1 | .. | 36·1 | 1,115 | 1,316 | 1,339 | .. | 3,770 | .. |
| Perseus, | F.S. | 20½ | 33 | 54 | .. | 27 | 266 | 24·2 | 214·8 | 0·75 | .. | .. | .. | .. | 52·5 | 992 | 1,263 | 1,268 | .. | 3,523 | 9·3 |
| | R.S. | | | | | | 251 | 25·1 | 197 | .. | .. | .. | .. | .. | 42·9 | 707 | 987 | 942 | .. | 2,636 | .. |
| Blanche, | F.D. | 21 | 31 | 45 | .. | 24 | 148 | 25·5 | 190·1 | 0·79 | 56·5 | 25·4 | 13·5 | .. | 39·8 | 451 | 442 | 497 | .. | 1,459 | 5·8 |
| | N.D. | | | | | | 147 | 25·8 | 197·6 | .. | 38·7 | 18·7 | 10·1 | .. | 27·4 | 273 | 287 | 325 | .. | 885 | .. |
| Dove, | F.D. | 19½ | 30 | 34 | 34 | 18 | 206 | 25 | 385·4 | 0·75 | 77·3 | 45·5 | 17·7 | 17·2 | 47·4 | 808 | 1,127 | 563 | 547 | 3,045 | 8·7 |
| Snapper, | F.D. | 19½ | 28½ | 43 | .. | 18 | 180 | 25 | 370·6 | 0·58 | 64·5 | 28·7 | 14·6 | .. | 39·7 | 635 | 592 | 715 | .. | 1,942 | 8·6 |

## Table XXV.—Results of Trials of Marine Engines (Twin Screw)—*continued.*

### MERCHANT SHIPS.

| | Screws. | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Victoria, T | T.S. | 41 | 67 | 106 | .. | 63 | 149 | 24 | 78 | 0·78 | 62·6 | 27·8 | .. | 11·16 | 32·8 | 2,214 | 2,470 | .. | 2,494 | 7,178 | 8·6 |
| Tmlipa, T | S.S. | 40 | 64 | 92 | .. | 60 | 140 | 23 | 77 | 0·73 | 55·1 | 21·4 | .. | 11·45 | 32·2 | 1,615 | 1,606 | .. | 1,775 | 4,996 | 7·2 |
| Saxonia, Q | T.S. | 29 | 41½ | 59 | 84 | 54 | 192 | 25·3 | 78 | .. | 73·1 | 41 | 20·9 | 12·08 | 39·0 | 985 | 1,158 | 1,202 | 1,420 | 4,765 | .. |
| Cazengo, T | S.S. | 32 | 48 | 80 | .. | 48 | 145 | .. | 71 | 0·65 | 68·1 | 32 | .. | 11·0 | 33·6 | 943 | 994 | .. | 947 | 2,884 | 9·5 |
| Othello, T | S.S. | 24¾ | 40 | 70 | .. | 48 | 200 | 28 | 71 | 0·59 | 88·5 | 36·8 | .. | 13·0 | 36·7 | 733 | 794 | .. | 903 | 2,430 | 15·2 |
| P. Grge, T | T.S. | 26 | 40 | 45 | 45 | 30 | 177 | 27 | 177 | 0·69 | 57·5 | 30·3 | 15·4 | 15·3 | 36·9 | 818 | 1,019 | 682 | 630 | 3,149 | 8·9 |
| Ohio, Q . | S.S. | 22½ | 32 | 45 | 64 | 42 | 200 | 27½ | 67·5 | 0·64 | 63·5 | 28·5 | 20·4 | 11 | 39·0 | 362 | 328 | 464 | 507 | 1,661 | 12·7 |
| P. Edwrd, T | T.S. | 19 | 30 | 48 | .. | 24 | 180 | 25 | 192·3 | 0·68 | 81 | 33·5 | .. | 13·7 | 39·6 | 536 | 554 | .. | 577 | 1,667 | 9·4 |
| Castro, T . | S.S. | 19 | 33 | 56 | .. | 36 | 200 | 25 | 91·5 | 0·59 | 87·8 | 33·5 | .. | 10·9 | 32·5 | 417 | 470 | .. | 445 | 1,333 | 14·7 |
| Colchstr, T | T.S. | 14½ | 23 | 27 | 27 | 33 | 180 | 26·5 | 145 | 0·75 | 83 | 38·0 | 18·4 | 18·2 | 44·6 | 331 | 382 | 255 | 252 | 1,220 | 9·3 |
| Molopo, TQ | S.S. | 13 | 22 | 36 | .. | 24 | 200 | 23·6 | 112 | 0·60 | 96·7 | 38·7 | .. | 13·1 | 40·0 | 174 | 199 | .. | 181 | 554 | 12·8 |
| Polo, Q . | S.S. | 12¾ | 18½ | 27 | 39 | 24 | 200 | 25 | 98 | 0·62 | 75 | 34·3 | 15·3 | 11·85 | 34·8 | 113 | 110 | 104 | 168 | 495 | 15·1 |
| Ptn, Q . . | S.S. | 24 | 34½ | 49 | 71 | 48 | 220 | 28·0 | 65 | .. | 80·3 | 32·1 | 16·0 | 9·12 | 33·5 | 572 | 476 | 472 | 568 | 2,088 | .. |
| Vsr, Q . . | S.S. | 26½ | 37½ | 59 | 77 | 57 | 230 | 27·5 | 75 | .. | 80·6 | 44·7 | 18·4 | 10·6 | 41·3 | 964 | 1,068 | 1,059 | 1,068 | 4,159 | .. |
| Tk, Q . . | S.S. | 20 | 28 | 39½ | 57 | 33 | 212 | 25·0 | 140 | 0·63 | 73·5 | 37·0 | 19·8 | 10·0 | 40·0 | 540 | 532 | 565 | 596 | 2,233 | 12·9 |
| Hl, Q . . | S.S. | 22½ | 31½ | 45 | 66 | 51 | 215 | 25·5 | 76 | 0·57 | 74·0 | 35·5 | 18·3 | 8·3 | 33·3 | 570 | 542 | 566 | 550 | 2,228 | 15·1 |
| Lru, T . . | S.S. | 22 | 35 | .. | 53 | 33 | 205 | 26·0 | 114 | .. | 81·2 | 33·2 | .. | 9·5 | 31·6 | 350 | 400 | .. | 330 | 1,080 | .. |
| Rw, T . . | S.S. | 25 | 40 | .. | 68 | 45 | 180 | 27·2 | 61 | .. | 68·7 | 27·1 | .. | 10·15 | 30·6 | 500 | 502 | .. | 545 | 1,547 | .. |

referred mean pressure is $Rp_m$, the mean pressures in the various cylinders are as follows:—

$$\text{L.P. mean pressure} = \frac{Rp_m}{N}.$$

$$\text{M.P.} \quad ,, \quad = \frac{Rp_m}{N} \times \frac{\text{L.P. area}}{\text{M.P. area}}.$$

$$\text{H.P.} \quad ,, \quad = \frac{Rp_m}{N} \times \frac{\text{L.P. area}}{\text{H.P. area}}.$$

## PISTON SPEED.

Usually reckoned in feet per minute.

Mean value = 2 × stroke in feet × revolutions per minute.

Experience with Naval machinery has shown that the largest pistons (*i.e.* up to 100 inches diameter) may be safely run at speeds up to 850 and 1000 feet per minute.

With the lighter machinery of torpedo boats and destroyers speeds of 1000 to 1200 feet per minute are quite satisfactory.

The standard practice in cargo steamers is very fairly represented by the expression $144 \sqrt[3]{S}$, where S is the stroke in inches. (North-East Coast rule—*see* page 5).

In large passenger and mail steamers the speed is now commonly 700 to 950 feet per minute on service, or, $220 \sqrt[3]{S}$.

In the case of paddle engines, and especially of oscillating paddle engines, the available range of piston speeds is very closely limited by the other conditions of the case, and has not often in practice been exceeded,—

| | | |
|---|---|---|
| For vertical oscillating engines, . . | 450 feet per min. | or $110 \sqrt[3]{S}$. |
| ,, inclined ,, . . | 520 ,, | or $118 \sqrt[3]{S}$. |
| ,, diagonal engines with connecting rods, | 550 to 700 ,, | or $130 \sqrt[3]{S}$. |

As the powers obtained from a given weight and bulk of engine vary almost directly as the revolutions, the efforts of engineers are constantly directed to obtain higher rates, without undue increase of wear and tear, and there is therefore every probability of advance in this direction.

By increasing piston speed a perceptible reduction can also be made in the percentage of heat lost by radiation.

There are two ways of getting increased piston speed, viz.:—

(1) Increase length of stroke and let revolutions remain same;

(2) Increase revolutions and let length of stroke remain same; but there is this important difference in the result obtained,—

In the first case, the cost, bulk, and weight of engines all increase with the piston speed; whilst in the second case there is practically no increase under any of these heads; and it is therefore in this second direction that the designer must go when limited as regards space and weight.

It must not be forgotten, however, that, for a given cylinder capacity, the long stroke and small diameter cylinder has a distinct advantage over its rival, in its smaller piston area, and consequently reduced loads and stresses, besides smaller *volume* clearances.

The chief limits to increase of piston speed (and of revolutions) are,—the size of ports and consequent size and weight of valves, and the inertia of reciprocating parts.

## REVOLUTIONS.

In the case of paddle engines, the revolutions are, of course, strictly governed by the diameter of wheel and speed of vessel; increased power must therefore be obtained either by increased stroke, pressure, or diameter of cylinders.

**Rate of Revolution** at full speed varies roughly as $\sqrt{\text{N.H.P.}}$ or $\sqrt{\frac{D \times S}{X}}$ *v.* Rule 7.

**Rule 34.** Revs. per min. $= Q \div \sqrt{\text{N.H.P.}}$

For ordinary merchant steamer, $Q = 1200$.
For express steamships, $Q = 1800$.
For naval and very fast expresses, $Q = 2250$.

**Rule 34a.** For paddle wheelers. Revs. per min. $= Q \div \sqrt[3]{\text{N.H.P.}}$
$Q = 280$.

For service on home waters and short runs, express paddle steamers of all powers with diagonal direct-driven engines run at about 50 revolutions per minute.

The following Table gives the maximum numbers of revolutions per minute usual in the classes of vessels named :—

### Table XXVI.—Revolutions, Rates of, in Practice.

| Class of Ship. | Revolutions. | Horse-Power of Main Engines. |
|---|---|---|
| Battleships, Reciprocators | 110 to 105 | 6000 to 9000 each engine |
| Do. Turbines | 330 to 380 | 27,000 to 23,000 total S.H.P. |
| First Class Cruisers, Recip. | 109 to 135 | 8500 to 15,000 each engine |
| Do. do. Turbines | 300 to 260 | 44,000 to 70,000 total S.H.P. |
| Second Class Cruisers, Recip. | 140 to 180 | 5000 to 6500 each engine |
| Do. do. Turbines | 500 to 480 | 22,000 to 27,000 total S.H.P. |
| Third Class Cruisers, Recip. | 225 to 245 | 3600 to 5000 each engine |
| Do. do. Turbines | 490 to 500 | 14,000 to 18,000 total S.H.P. |
| Scouts, Recip. | 210 to 202 | 7000 to 8500 each engine |
| Torpedo Boat Destroyers, Do. | 380 to 410 | 3000 to 4000 do. |
| Do. do. do. Turbines | 940 to 760 | 7500 to 16,250 total S.H.P. |
| Large Express Steamers, Recip. | 75 to 95 | 8000 to 16,000 each engine |
| Do. do. do. Turbines | 185 to 165 | 24,000 to 65,000 total S.H.P. |
| Medium do. do. Recip. | 75 to 86 | 8000 to 3000 each engine |
| Do. do. do. Turbines | 275 to 325 | 18,000 to 12,000 total S.H.P. |
| Channel do. do. Recip. | 140 to 200 | 2500 to 4000 each engine |
| Do. do. do. Turbines | 456 to 750 | 14,500 to 7000 total S.H.P. |
| Ordinary Cargo do. Recip. | 60 to 90 | 800 to 2500 each engine |
| Paddle Wheel do. do. | 40 to 55 | 10,000 to 600 |
| Geared Turbines | 3000 to 3500 | Any power |

### Table XXVII.—Relation between Stroke and Revolutions, Cargo Steamers.

| Stroke in inches. | Revolutions per minute. | Stroke in inches. | Revolutions per minute. | Stroke in inches. | Revolutions per minute. |
|---|---|---|---|---|---|
| 18 | 125 | 33 | 84 | 48 | 65 |
| 21 | 113 | 36 | 79 | 51 | 63 |
| 24 | 103 | 39 | 75 | 54 | 61 |
| 27 | 96 | 42 | 71 | 57 | 58 |
| 30 | 89 | 45 | 68 | 60 | 56 |

## STROKE OF PISTON

The following are the lengths of stroke usual in the various classes of vessels named :—

### Table XXVIII.—Stroke of Piston.

| | |
|---|---|
| Battle Ships and First Class Cruisers, . . | 45 to 51 inches. |
| Second Class Cruisers, . . . . . . | 36 to 42 ,, |
| Scouts and Third Class Cruisers, . . . | 24 to 30 ,, |
| Torpedo Gunboats and Destroyers, . . . | 18 to 21 ,, |
| Torpedo Boats, . . . . . . . | 9 to 18 ,, |
| Large Mail Steamers, . . . . . | 60 to 78 ,, |
| Fast Passenger Steamers, . . . . | 30 to 48 ,, |
| Ordinary Merchant Steamers, . . . | 18 to 60 ,, |
| Paddle Steamers, . . . . . . | 30 to 102 ,, |

Engines of the overhead beam type, used in paddle vessels, have sometimes had strokes of 10 feet and even 12 feet.

In ordinary cargo steamers the length of stroke is usually about ·65 × diameter of L.P. cylinder. (*See* Table I., page 5).

## TO CALCULATE DIAMETER OF CYLINDER FOR A GIVEN POWER.

The following rules apply to L.P. cylinders only,—the sizes of M.P. and H.P. cylinders being supposed to be fixed from those of the L.P. by the rules given in section on "Ratios of Cylinders," pages 57–65.

For rapid calculations, use the formula for Estimated Horse Power, Rule 1 ; the result will usually be within 5 per cent.

**Rule 1.** $$\text{E.H.P.} = \frac{D^2 \times \sqrt{p} \times R \times S}{Z}$$

where D = diameter of L.P. cylinder in inches.
$p$ = absolute boiler pressure.
R = revolutions per minute.
S = stroke in feet.
Z a factor (*v.* page 3).

If the size of cylinder for a given **Indicated Horse-power** is required, let S stand for piston speed in feet per minute, and $Rp_m$ for the referred mean pressure in pounds per square inch (calculated by rules given on pages 67, 68); then,—

**Rule 35.** $$\textbf{Area of L.P. piston} = \frac{\text{I.H.P.} \times 33{,}000}{Rp_m \times S};$$

or, if piston speed is required,—

**Rule 36.** $$\textbf{Piston speed} = \frac{\text{I.H.P.} \times 33{,}000}{\text{L.P. area} \times Rp_m};$$

or, again, if referred mean pressure is required,—

**Rule 37.** $$\text{Referred mean pressure } Rp_m = \frac{\text{I.H.P.} \times 33{,}000}{\text{L.P. area} \times S};$$

and lastly, if I.H.P. is required,—

**Rule 38.** $$\text{I.H.P.} = \frac{\text{L.P. area} \times S \times Rp_m}{33{,}000}.$$

## CYLINDER PORTS, PIPES, AND PASSAGES.

In fixing the sizes of cylinder ports, &c., it is necessary—as in so many other cases—to discover the best compromise. By increasing the port areas, for instance, "wire-drawing" is diminished, a freer exhaust is obtained, and the resulting indicator diagram is fuller; but, at the same time, the size and weight of cylinder, of slide-valve, and of valve-gear, as also the clearance volume, are increased, the loss from which may possibly balance the gain.

The following figures give speeds of steam usual in good triple and quadruple engines; but it must be understood that in dealing with very high piston speeds (say over 900 feet per minute), it is not always either possible or advisable to give such large areas:—

## Table XXIX.—Speeds of Steam-flow through Ports, &c.

Main steam pipe, 8100 ft. per minute; then—

$$\text{Diameter} = \frac{\text{Dia. of H.P. cyl.}}{90} \times \sqrt{\text{Mean piston speed.}}$$

| | | | |
|---|---|---|---|
| Mean of maximum valve openings, | H.P. — 7,500 | ft. per min. | |
| | M.P. — 9,000 | ,, | |
| | L.P. —12,000 | ,, | |
| Ports (during exhaust), | H.P. — 5,800 | ,, | Nearly equivalent to 40, 50, and 60 c. ft. of cylinder per sq. inch of port per minute. |
| | M.P. — 7,200 | ,, | |
| | L.P. — 8,600 | ,, | |
| Exhaust pipe or passage from one cylinder to next or to condenser, | H.P. — 4,500 | ,, | |
| | M.P. — 5,500 | ,, | |
| | L.P. — 7,000 | ,, | |
| Ports (during exhaust) in light, high-speed engines, | H.P. — 5,800 | ,, | Nearly equivalent to 40, 60, and 80 c. ft. of cylinder per sq. inch of port per minute. |
| | M.P. — 8,600 | ,, | |
| | L.P. —11,500 | ,, | |

For **Two-stage Compound** engines use the above figures, only omitting those referring to M.P. cylinders, and for quadruples the mean speeds between H.P. and M.P. and M.P. and L.P.

The following Table is based on the above figures, and gives the proportions at a glance,—where A is the area of the cylinder :—

## Table XXX.—Ratios for Ports, Pipes, &c., of Cylinders.

| Piston speed, feet per minute. | H.P.A. ÷ Area of main steam. | H.P.A. ÷ Mean area of H.P. max. openings. | M.P.A. ÷ Mean area of M.P. max. openings. | L.P.A. ÷ Mean area of L.P. max. openings. | H.P.A. ÷ Area of H.P. ports. | M.P.A. ÷ Area of M.P. ports. | L.P.A. ÷ Area of L.P. ports. | H.P.A. ÷ Area of pipe from H.P. to M.P. | M.P.A. ÷ Area of pipe from M.P. to L.P. | L.P.A. ÷ Area of main eduction. |
|---|---|---|---|---|---|---|---|---|---|---|
| 200 | 40·0 | 37·5 | 45·0 | 60·0 | 29·0 | 36·0 | 43·0 | 22·5 | 27·5 | 35·0 |
| 250 | 32·0 | 30·0 | 36·0 | 48·0 | 23·2 | 28·8 | 34·4 | 18·0 | 22·0 | 28·0 |
| 300 | 26·6 | 25·0 | 30·0 | 40·0 | 19·3 | 24·0 | 28·6 | 15·0 | 18·3 | 23·3 |
| 350 | 22·8 | 21·4 | 25·7 | 34·3 | 16·6 | 20·6 | 24·6 | 12·8 | 15·7 | 20·0 |
| 400 | 20·0 | 18·7 | 22·5 | 30·0 | 14·5 | 18·0 | 21·5 | 11·2 | 13·7 | 17·5 |
| 450 | 17·7 | 16·6 | 20·0 | 26·6 | 12·9 | 16·0 | 19·1 | 10·0 | 12·2 | 15·5 |
| 500 | 16·0 | 15·0 | 18·0 | 24·0 | 11·6 | 14·4 | 17·2 | 9·0 | 11·0 | 14·0 |
| 550 | 14·5 | 13·6 | 16·3 | 21·8 | 10·5 | 13·1 | 15·6 | 8·2 | 10·0 | 12·7 |
| 600 | 13·3 | 12·5 | 15·0 | 20·0 | 9·6 | 12·0 | 14·3 | 7·5 | 9·1 | 11·6 |
| 650 | 12·3 | 11·5 | 13·9 | 18·4 | 8·9 | 11·1 | 13·2 | 6·9 | 8·4 | 10·7 |
| 700 | 11·4 | 10·7 | 12·8 | 17·1 | 8·3 | 10·3 | 12·3 | 6·4 | 7·8 | 10·0 |
| 750 | 10·6 | 10·0 | 12·0 | 16·0 | 7·7 | 9·6 | 11·5 | 6·0 | 7·3 | 9·3 |
| 800 | 10·0 | 9·4 | 11·2 | 15·0 | 7·2 | 9·0 | 10·7 | 5·6 | 6·9 | 8·7 |
| 850 | 9·4 | 8·8 | 10·6 | 14·1 | 6·8 | 8·5 | 10·1 | 5·3 | 6·5 | 8·2 |
| 900 | 8·9 | 8·3 | 10·0 | 13·3 | 6·4 | 8·0 | 9·5 | 5·0 | 6·1 | 7·8 |
| 950 | 8·4 | 7·9 | 9·5 | 12·6 | 6·1 | 7·6 | 9·0 | 4·7 | 5·8 | 7·4 |
| 1000 | 8·0 | 7·5 | 9·0 | 12·0 | 5·8 | 7·2 | 8·6 | 4·5 | 5·5 | 7·0 |

To determine the area of any port, pipe, passage, &c., for any given speed of piston, —Divide the area of the cylinder by the number found in the proper column opposite the given piston speed, and quotient will be area required.

## GENERAL REMARKS ON STRENGTHS, &c.

The dimensions and proportions given for the various details of engines, in this and succeeding sections, are in all cases such as experience has shown to give satisfactory results, both as regards stiffness, strength, and durability. Where weight is not an important consideration the various parts may be made heavier, but with good material this is unnecessary ; on the other hand, they may be, and are sometimes, made lighter,—as in very high-speed Naval vessels,—but in such cases the increased risk of breakdown, and wear and tear when working at full power, are understood, and accepted as the price paid for reduction in weight. Such ships, however, do not run often or for long periods at full power.

## CYLINDER BARRELS.

Cylinder barrels should be made of good sound cast-iron, at least twice melted, and of the thicknesses given by the following rules :—

**Rule 39.** $T = \frac{D}{6500}(p + 50) + \cdot 2$ ; for cylinders fitted with liners.

**Rule 39a.** $T = \frac{D}{6500}(p + 50) + \cdot 4$ ; for cylinders without liners.

Where T is thickness of barrel, and D diameter of cylinder, both in inches ; and $p$ is maximum pressure in cylinder,—the values of which are assumed to be as follows :—

For H.P. cylinders of Triple and Quadruple engines, $p =$ boiler pressure.
H.P. ,, Compound ,, $p =$ boiler pressure $+ 20$.
M.P. ,, Triple ,, $p = \cdot 6$ boiler pressure.
$\text{M.P.}_1$ ,, Quadruple ,, $p = \cdot 7 \times$ ,,
$\text{M.P.}_2$ ,, ,, ,, $p = \cdot 45 \times$ ,,
L.P. ,, Compound ,, $p = \cdot 5 \times (\text{boiler pressure} + 20)$.
L.P. ,, Triple ,, $p = \cdot 37 \times$ boiler pressure.
L.P. ,, Quadruple ,, $p = \cdot 3 \times$ ,,

It is also assumed, as previously stated, that compound engines work at pressure between 70 lbs. and 120 lbs. ; Triple engines between 120 lbs. and 180 lbs. ; and Quadruple engines between 180 lbs. and 240 lbs.

For working pressure of 160 lbs. the formula may be reduced to the forms given at the heads of the columns in the following Table, which is calculated in accordance with them, for 180 lbs. :—

## Table XXXI.—Cylinder Barrels of Triple Engines (180 lbs.).

| Dia. of Cylinder in ins. | Thickness in inches, WITH LINERS. | | | | | | Thick. in inches, WITHOUT LINERS. | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | H.P. $\frac{D}{28·5}+·2$ | | M.P. $\frac{D}{41}+·2$ | | L.P. $\frac{D}{55}+·2$ | | H.P $\frac{D}{28·5}+·4$ | | M.P. $\frac{D}{41}+·4$ | | L.P. $\frac{D}{55}+·4$ | |
| 10 | | | | | | | ·81 | $\frac{13}{16}$ | ·75 | $\frac{3}{4}$ | ·58 | $\frac{5}{8}$ |
| 15 | ·90 | $\frac{15}{16}$ | | | | | ·95 | $\frac{15}{16}$ | ·81 | $\frac{13}{16}$ | ·67 | $\frac{11}{16}$ |
| 20 | 1·07 | $1\frac{1}{16}$ | | | | | 1·10 | $1\frac{1}{8}$ | ·89 | $\frac{15}{16}$ | ·76 | $\frac{13}{16}$ |
| 25 | 1·25 | $1\frac{1}{4}$ | ·81 | $\frac{13}{16}$ | | | 1·27 | $1\frac{1}{4}$ | 1·01 | 1 | ·85 | $\frac{7}{8}$ |
| 30 | 1·42 | $1\frac{7}{16}$ | ·93 | $\frac{15}{16}$ | | | 1·45 | $1\frac{7}{16}$ | 1·13 | $1\frac{1}{8}$ | ·94 | 1 |
| 35 | 1·60 | $1\frac{5}{8}$ | 1·05 | $1\frac{1}{16}$ | ·83 | $\frac{13}{16}$ | 1·62 | $1\frac{5}{8}$ | 1·25 | $1\frac{1}{4}$ | 1·03 | $1\frac{1}{16}$ |
| 40 | 1·77 | $1\frac{3}{4}$ | 1·17 | $1\frac{3}{16}$ | ·93 | $\frac{15}{16}$ | 1·80 | $1\frac{13}{16}$ | 1·37 | $1\frac{3}{8}$ | 1·13 | $1\frac{1}{8}$ |
| 45 | 1·95 | $1\frac{15}{16}$ | 1·29 | $1\frac{5}{16}$ | 1·02 | $1\frac{1}{16}$ | 1·97 | 2 | 1·49 | $1\frac{1}{2}$ | 1·22 | $1\frac{1}{4}$ |
| 50 | 2·12 | $2\frac{1}{8}$ | 1·42 | $1\frac{7}{16}$ | 1·11 | $1\frac{1}{8}$ | 2·15 | $2\frac{1}{8}$ | 1·62 | $1\frac{5}{8}$ | 1·31 | $1\frac{5}{16}$ |
| 55 | 2·30 | $2\frac{5}{16}$ | 1·54 | $1\frac{9}{16}$ | 1·20 | $1\frac{1}{4}$ | 2·32 | $2\frac{5}{16}$ | 1·74 | $1\frac{3}{4}$ | 1·40 | $1\frac{7}{16}$ |
| 60 | | | 1·66 | $1\frac{11}{16}$ | 1·29 | $1\frac{5}{16}$ | 2·50 | $2\frac{1}{2}$ | 1·86 | $1\frac{7}{8}$ | 1·49 | $1\frac{1}{2}$ |
| 65 | | | 1·78 | $1\frac{13}{16}$ | 1·38 | $1\frac{3}{8}$ | | | 1·98 | 2 | 1·58 | $1\frac{9}{16}$ |
| 70 | | | 1·90 | $1\frac{15}{16}$ | 1·47 | $1\frac{1}{2}$ | | | 2·10 | $2\frac{1}{8}$ | 1·67 | $1\frac{11}{16}$ |
| 75 | | | 2·03 | $2\frac{1}{16}$ | 1·56 | $1\frac{9}{16}$ | | | 2·23 | $2\frac{1}{4}$ | 1·76 | $1\frac{3}{4}$ |
| 80 | | | 2·15 | $2\frac{3}{16}$ | 1·65 | $1\frac{11}{16}$ | | | 2·35 | $2\frac{3}{8}$ | 1·85 | $1\frac{7}{8}$ |
| 85 | | | | | 1·74 | $1\frac{3}{4}$ | | | | | 1·94 | $1\frac{15}{16}$ |
| 90 | | | | | 1·83 | $1\frac{7}{8}$ | | | | | 2·03 | $2\frac{1}{16}$ |
| 95 | | | | | 1·93 | $1\frac{15}{16}$ | | | | | 2·13 | $2\frac{1}{8}$ |
| 100 | | | | | 2·02 | $2\frac{1}{16}$ | | | | | 2·22 | $2\frac{1}{4}$ |
| 105 | | | | | 2·11 | $2\frac{1}{8}$ | | | | | 2·31 | $2\frac{5}{16}$ |
| 110 | | | | | 2·20 | $2\frac{1}{4}$ | | | | | 2·40 | $2\frac{7}{16}$ |

The following general rule will be useful where pressures of over 200 lbs. per square inch are in question, as it has a wider range than those given above :—

Rule 40. —For the thickness of any cylinder of a compound, triple, or quadruple engine.

$$\text{Thickness} = \frac{D}{6400}(p+80) \times \frac{1}{\sqrt{r}} + \begin{cases} ·2 \text{ with liner.} \\ ·4 \text{ without liner.} \end{cases}$$

where D is diameter of cylinder in inches, $p$ working pressure at engines, and $r$ has the values given below.

| Working Pressure. | Values of $r$. | | | | |
|---|---|---|---|---|---|
| | H.P. | I.P. Triple. | 1st I.P. Quadruple. | 2nd I.P. Quadruple. | L.P. |
| 100 lbs. | 1 | ... | ... | ... | 2·5 |
| 160 ,, | 1 | 2·3 | ... | ... | 4 |
| 200 ,, | 1 | 2·5 | 1·7 | 3 | 5·55 |
| 250 ,, | 1 | 2·75 | 1·75 | 3·45 | 7·5 |

## Table XXXII.—Cylinder Barrels of 2-Stage Compound Engines (100 lbs.).

| Dia. of Cylinder in ins. | Thickness in inches, WITH LINERS. |  |  |  | Thick. in inches, WITHOUT LINERS. |  |  |  |
|---|---|---|---|---|---|---|---|---|
|  | H.P. $\frac{D}{35{\cdot}3}+{\cdot}2$ |  | L.P. $\frac{D}{55}+{\cdot}2$ |  | H.P. $\frac{D}{35{\cdot}3}+{\cdot}4$ |  | L.P. $\frac{D}{55}+{\cdot}4$ |  |
| 10 |  |  |  |  | ·68 | 11/16 | ·58 | ⅝ |
| 15 |  |  |  |  | ·82 | 13/16 | ·67 | 11/16 |
| 20 | ·76 | ¾ |  |  | ·96 | 1 | ·76 | 13/16 |
| 25 | ·91 | 15/16 |  |  | 1·11 | 1⅛ | ·85 | ⅞ |
| 30 | 1·05 | 1 1/16 |  |  | 1·25 | 1¼ | ·94 | 1 |
| 35 | 1·19 | 1 3/16 | ·83 | 13/16 | 1·39 | 1⅜ | 1·03 | 1 1/16 |
| 40 | 1·33 | 1 5/16 | ·93 | 15/16 | 1·53 | 1½ | 1·13 | 1⅛ |
| 45 | 1·47 | 1½ | 1·02 | 1 1/16 | 1·67 | 1 11/16 | 1·22 | 1¼ |
| 50 | 1·61 | 1⅝ | 1·11 | 1⅛ | 1·81 | 1 13/16 | 1·31 | 1 5/16 |
| 55 | 1·75 | 1¾ | 1·20 | 1¼ | 1·95 | 1 15/16 | 1·40 | 1 7/16 |
| 60 | 1·90 | 1⅞ | 1·29 | 1 5/16 | 2·10 | 2⅛ | 1·49 | 1½ |
| 65 | 2·04 | 2 1/16 | 1·38 | 1⅜ | 2·24 | 2¼ | 1·58 | 1 9/16 |
| 70 |  |  | 1·47 | 1½ |  |  | 1·67 | 1 11/16 |
| 75 |  |  | 1·56 | 1 9/16 |  |  | 1·76 | 1¾ |
| 80 |  |  | 1·65 | 1 11/16 |  |  | 1·85 | 1⅞ |
| 85 |  |  | 1·74 | 1¾ |  |  | 1·94 | 1 15/16 |
| 90 |  |  | 1·83 | 1⅞ |  |  | 2·03 | 2 1/16 |
| 95 |  |  | 1·93 | 1 15/16 |  |  | 2·13 | 2⅛ |
| 100 |  |  | 2·02 | 2 1/16 |  |  | 2·22 | 2¼ |
| 105 |  |  | 2·11 | 2⅛ |  |  | 2·31 | 2 5/16 |
| 110 |  |  | 2·20 | 2¼ |  |  | 2·40 | 2 7/16 |

For the very light machinery of Torpedo boats and Destroyers, the thicknesses of cylinders for triple engines working at pressures of 220 to 250 lbs. may be as shown in the following Table :—

## Table XXXIII.—Cylinder Barrels of Torpedo Boats and Destroyers.

| Diameter of cylinder in inches. | H.P. |  | M.P. |  | L.P. |  |
|---|---|---|---|---|---|---|
|  | Thickness in inches. $\frac{D}{40}+{\cdot}34.$ |  | Thickness in inches. $\frac{D}{80}+{\cdot}4.$ |  | Thickness in inches. $\frac{D}{120}+{\cdot}42.$ |  |
| 10 | ·59 | 9/16 | ... | ... | ... | ... |
| 15 | ·71 | 11/16 | ·58 | 9/16 | ... | ... |
| 20 | ·84 | 13/16 | ·64 | ⅝ | ·58 | 9/16 |
| 25 | ·96 | 15/16 | ·71 | 11/16 | ·62 | ⅝ |
| 30 | 1·09 | 1 1/16 | ·77 | ¾ | ·66 | 11/16 |
| 35 | 1·21 | 1 3/16 | ·83 | 13/16 | ·71 | 11/16 |
| 40 |  |  | ·90 | ⅞ | ·75 | ¾ |
| 45 |  |  | ·96 | 15/16 | ·79 | 13/16 |
| 50 |  |  |  |  | ·84 | 13/16 |
| 55 |  |  |  |  | ·88 | ⅞ |

The mixture of iron for cylinders of this type must, of course, be of a very special character, and should contain a good proportion of best cold-blast iron.

For thicknesses of cylinder barrels for oscillating engines see page 99.

The barrels of all cylinders, but especially the L.P., are improved by the addition of external stiffening ribs or rings; these may have a thickness of 1·5 × thickness of barrel, and may stand ·75 × thickness of barrel above the surface, whilst they may be pitched about 12 × thickness of barrel apart.

## CYLINDER LINERS.

Cast-iron cylinder liners should be of thickness given by the rule,—

**Rule 41.**

$$T = \cdot 8 \left\{ \frac{D}{6500}(p+50) \right\} + \cdot 35$$

where T is thickness, and D diameter, both in inches, and $p$ has the same values as given in section on cylinder barrels, above.

The following Table is calculated by means of the above formula, for liners of triple engines working at 160 lbs. pressure :—

Table XXXIV.—Cast-iron Cylinder Liners (180 lbs.).

| Diameter in inches. | Thickness in inches. | | | | | | Diameter in inches. | Thickness in inches. | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | H.P. | | M.P. | | L.P. | | | H.P. | | M.P. | | L.P. | |
| 10 | ... | ... | .. | ... | ... | ... | 65 | ... | ... | 1·62 | 1 5/8 | 1·29 | 1 5/16 |
| 15 | ... | ... | ... | ... | ... | ... | 70 | ... | ... | 1·71 | 1 11/16 | 1·36 | 1 3/8 |
| 20 | 0·91 | 15/16 | ... | ... | ... | ... | 75 | ... | ... | 1·81 | 1 13/16 | 1·44 | 1 7/16 |
| 25 | 1·05 | 1 1/16 | 0·83 | 13/16 | ... | ... | 80 | ... | ... | 1·91 | 1 15/16 | 1·51 | 1 1/2 |
| 30 | 1·19 | 1 3/16 | 0·93 | 15/16 | ... | ... | 85 | ... | ... | ... | ... | 1·59 | 1 9/16 |
| 35 | 1·33 | 1 5/16 | 1·03 | 1 | 0·85 | 7/8 | 90 | ... | ... | ... | ... | 1·66 | 1 5/8 |
| 40 | 1·47 | 1 1/2 | 1·13 | 1 1/8 | 0·92 | 15/16 | 95 | ... | ... | ... | ... | 1·73 | 1 3/4 |
| 45 | 1·61 | 1 5/8 | 1·22 | 1 1/4 | 1·00 | 1 | 100 | ... | ... | ... | ... | 1·80 | 1 13/16 |
| 50 | 1·75 | 1 3/4 | 1·32 | 1 5/16 | 1·08 | 1 1/16 | 105 | ... | ... | ... | ... | 1·88 | 1 7/8 |
| 55 | 1·89 | 1 7/8 | 1·42 | 1 7/16 | 1·15 | 1 1/8 | 110 | ... | ... | ... | ... | 1·95 | 1 15/16 |
| 60 | 2·03 | 2 | 1·52 | 1 1/2 | 1·22 | 1 1/4 | ... | ... | ... | ... | ... | ... | ... |

When of forged steel, cylinder liners should be of thickness given by

**Rule 42.**

$$T = \cdot 65 \left\{ \frac{D}{6500}(p+50) \right\} + \cdot 3$$

where all symbols have same meanings as above.

Table XXXV. is calculated by means of this formula, for liners of triple engines working at 180 lbs. pressure.

Table XXXV.—Forged steel Cylinder Liners (180 lbs.).

| Diam. of cylinder in inches. | Thickness in inches. | | | | | |
|---|---|---|---|---|---|---|
| | H.P. | | M.P. | | L.P. | |
| 20 | ·75 | ¾ | | | | |
| 25 | ·87 | ⅞ | | | | |
| 30 | ·98 | 1 | ·77 | $\frac{13}{16}$ | | |
| 35 | 1·10 | 1⅛ | ·85 | ⅞ | | |
| 40 | 1·21 | 1¼ | ·93 | $\frac{15}{16}$ | | |
| 45 | 1·33 | $1\frac{5}{16}$ | 1·01 | 1 | ·83 | ⅞ |
| 50 | 1·44 | $1\frac{7}{16}$ | 1·09 | 1⅛ | ·89 | $\frac{15}{16}$ |
| 55 | 1·55 | $1\frac{9}{16}$ | 1·17 | $1\frac{3}{16}$ | ·95 | 1 |
| 60 | 1·66 | $1\frac{11}{16}$ | 1·25 | 1¼ | 1·01 | $1\frac{1}{16}$ |
| 65 | | | 1·33 | $1\frac{5}{16}$ | 1·07 | $1\frac{1}{16}$ |
| 70 | | | 1·41 | $1\frac{7}{16}$ | 1·12 | 1⅛ |
| 75 | | | 1·49 | 1½ | 1·18 | $1\frac{3}{16}$ |
| 80 | | | 1·57 | $1\frac{9}{16}$ | 1·24 | 1¼ |
| 85 | | | | | 1·30 | $1\frac{5}{16}$ |
| 90 | | | | | 1·36 | 1⅜ |
| 95 | | | | | 1·42 | $1\frac{7}{16}$ |
| 100 | | | | | 1·48 | 1½ |
| 105 | | | | | 1·54 | $1\frac{9}{16}$ |
| 110 | | | | | 1·60 | 1⅝ |

For liners of horizontal cylinders the above thicknesses may be increased by $^1/_{16}$-inch.

For steel liners the piston packing rings should be of hard bronze, as it has been found to work more satisfactorily than cast iron with the forged steel.

For cylinder liners of engines working at pressures over 200 lbs. per square inch, substitute $\frac{D}{7000}(p + 80) \times \frac{1}{\sqrt{r}}$ for $\frac{D}{6500}(p + 50)$ in Rules 41 and 42.

## CYLINDER ENDS AND COVERS.

In most engines the cylinder end has, in addition to supporting the uniformly distributed load due to the steam pressure, to take the more or less locally applied pull of the frames or columns, and distribute it to the barrel; and the various ribs or webs must therefore be carefully arranged to effect this.

When the L.P. diameter is over 70 inches, it is desirable to make the end double, and this must of course be done for any size when steam jackets are required.

The double bottom adds considerably to the weight of a cylinder, as the inner metal must still be made strong enough to stand all local shock and strain; and it increases the risk of unsoundness in the casting.

In Naval work the double bottom is frequently dispensed with, on account of its weight; but owing to the use of the conical or "dished" steel piston, the cylinder end can also be dished, and is thus given such additional structural strength that L.P. cylinders are commonly made with single ends up to 90 inches diameter.

Table XXXVI. gives the thicknesses of single and double cylinder ends for triple engines working at 180 lbs. pressure, the following conditions being assumed:—

(*a*) The over-all depth or thickness of a double end is not less than five times that of the metal given in the table.

(*b*) In a single end, the total depth of the central ring (forming the hole for boring-bar) is not less than 5½ times the thickness of end given in the table.

(*c*) The number, strengths, and positions of webs, &c., are as customary in "average" practice for two, three, or four columns or frame attachments.

As much depends on (*c*) the Table and rules from which it is calculated must be used with judgment, and checked by the rule for flat surfaces given on page 90.

The rules from which Table XXXVI. is calculated are,—

**Rule 43.** Thickness of single cylinder end $= \cdot 85 \times f$.

**Rule 43a.** Thickness of double cylinder end $= \cdot 7 = f$,

where $f$ is thickness of barrel (columns 8, 10, and 12, Table XXXI.) $+ \cdot 25$ in.

## Table XXXVI.—Cylinder Ends of Triple Compound Engines.

| Diam. of cylin. in inches | H.P.—Thick. in inches. | | | | M.P.—Thick. in inches. | | | | L.P.—Thick. in inches. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Single end. | | Double end. | | Single end. | | Double end. | | Single end. | | Double end. | |
| 10 | ·85 | 7/8 | ·70 | | ·76 | 3/4 | ·63 | | ·70 | 3/4 | ·58 | |
| 15 | 1·00 | 1 | ·83 | | ·87 | 7/8 | ·71 | | ·78 | 13/16 | ·64 | |
| 20 | 1·15 | 1 3/16 | ·94 | | ·97 | 1 | ·80 | | ·86 | 7/8 | ·71 | |
| 25 | 1·29 | 1 5/16 | 1·06 | | 1·07 | 1 1/16 | ·88 | | ·94 | 15/16 | ·77 | |
| 30 | 1·44 | 1 7/16 | 1·19 | 1 3/16 | 1·17 | 1 3/16 | ·96 | | 1·01 | 1 | ·83 | |
| 35 | 1·59 | 1 9/16 | 1·31 | 1 5/16 | 1·27 | 1 1/4 | 1·05 | | 1·09 | 1 1/8 | ·89 | |
| 40 | 1·74 | 1 3/4 | 1·43 | 1 7/16 | 1·38 | 1 3/8 | 1·13 | 1 1/8 | 1·17 | 1 3/16 | ·96 | |
| 45 | 1·89 | 1 7/8 | 1·55 | 1 9/16 | 1·48 | 1 1/2 | 1·22 | 1 1/4 | 1·25 | 1 1/4 | 1·03 | |
| 50 | 2·04 | 2 1/16 | 1·67 | 1 11/16 | 1·59 | 1 5/8 | 1·31 | 1 5/16 | 1·32 | 1 5/16 | 1·09 | |
| 55 | 2·19 | 2 3/16 | 1·80 | 1 13/16 | 1·69 | 1 11/16 | 1·40 | 1 7/16 | 1·40 | 1 3/8 | 1·15 | |
| 60 | 2·34 | 2 5/16 | 1·92 | 1 15/16 | 1·79 | 1 13/16 | 1·48 | 1 1/2 | 1·48 | 1 1/2 | 1·22 | |
| 65 | | | | | 1·89 | 1 7/8 | 1·54 | 1 9/16 | 1·56 | 1 9/16 | 1·28 | 1 5/16 |
| 70 | | | | | 2·00 | 2 | 1·60 | 1 5/8 | 1·63 | 1 5/8 | 1·35 | 1 3/8 |
| 75 | | | | | 2·11 | 2 1/8 | 1·66 | 1 11/16 | 1·71 | 1 3/4 | 1·41 | 1 7/16 |
| 80 | | | | | 2·21 | 2 3/16 | 1·82 | 1 13/16 | 1·78 | 1 13/16 | 1·47 | 1 1/2 |
| 85 | | | | | | | | | 1·86 | 1 7/8 | 1·53 | 1 9/16 |
| 90 | | | | | | | | | 1·94 | 1 15/16 | 1·60 | 1 5/8 |
| 95 | | | | | | | | | 2·02 | 2 | 1·66 | 1 11/16 |
| 100 | | | | | | | | | 2·10 | 2 1/8 | 1·73 | 1 3/4 |
| 105 | | | | | | | | | 2·18 | 2 3/16 | 1·79 | 1 13/16 |
| 110 | | | | | | | | | 2·25 | 2 1/4 | 1·85 | 1 7/8 |

N.B.—The stiffening by webs and gussets must be made to suit the various initial pressures.

The practice of making the inner metal of double bottoms and covers a little heavier than the outer in large cylinders is reasonable, the inner metal may be made $\frac{1}{16}$-inch thicker than shown in Table, and the outer $\frac{1}{16}$-inch lighter than table.

Covers are now often made single and well ribbed both in cast iron and steel; the ribs of the larger covers have bulb or T edges. Single covers should be of thickness given by,—

**Rule 44.** **Thickness of single cylinder cover** $= \cdot 77 \times f$;

and for double covers,—

**Rule 44a.** **Thickness of double cylinder cover** $= \cdot 65 \times f$;

similar general conditions being assumed, as in the case of cylinder ends.

A sufficiently close approximation to these thicknesses, for triple engines working at 180 lbs. pressure, may be obtained by multiplying the thicknesses given in Table XXXVI. for cylinder ends by ·95.

In large covers and bottoms there should be no lack of strength in the central ring which surrounds the stuffing-box or manhole in the one case, and the boring hole in the other; it should have the same thickness as the metal of the cover or end, and good round corners formed by its junction with the metals of the bottom or cover, and of the various webs.

Very light cylinders, of the type referred to in Table XXXIII., are better made with loose covers for both ends, the thicknesses of which should be as follows :—

**Rule 45.** **Thickness of end covers** $= \cdot 5 \times f_1$;

where $f_1$ is the thickness of cylinder barrel (from Table XXXIII.) $+ \cdot 25$ inch.

They should be made of the same special iron as the barrels, and should each have from 6 to 12 deep stiffening ribs, the mean thickness of which (they should have $\frac{1}{16}$-inch taper for moulding) may be $\frac{1}{16}$-inch less than that of the cover.

## CYLINDER VALVE BOXES, PORTS, &c.

The casings, ports, passages, &c., for piston slide-valves should be made of thicknesses given by,—

**Rule 46.** **Thickness of piston valve casing** $= \cdot 8 \times f$;

and when flat slide-valves are used the thicknesses of casings may be,—

**Rule 47.** Thickness of valve casing $= \cdot 7 \times f$;

$f$ having, in both cases, the same value as above.

It will be noted that the latter value ($\cdot 7 \times f$) is the same as for metal of double ends, the thicknesses of which (for triple engines working at 180 lbs. pressure) are given in Table XXXVI.

It must be remembered that in the case of valve casings, as in that of cylinder ends, a very great deal depends on the number, size, and arrangement of stiffening webs, and the following rule should therefore be used in conjunction with the Table :—

**Rule for flat surfaces.**—All flat surfaces should be stiffened by webs or stays of some form, whose distance apart should not exceed,—

**Rule 48.** Pitch in inches $= \sqrt{\dfrac{t^2 \times C}{p}}$;

where $t$ is the thickness in 16ths of an inch, $p$ the pressure in pounds per square inch on the surface, and C equals 54 for cast iron, and 120 for cast steel. *See* also Table CXXXIX.

These webs should be of the same thickness as the flat surface, and their depth at least 2·5 times the thickness.

When flat slide-valves are used the thickness of the metal which separates the steam ports from the steam chest or receiver, and from the exhaust port, should be in accordance with the following rule,—it being understood that the ports of large cylinders are so divided by webs that no section or compartment exceeds 20 inches in width :—

**Rule 49.** Thickness of steam passage metal $= \cdot 65 \times f$,

where $f$ has the same value as before.

These thicknesses (for triple engines working at 180 lbs. pressure) may be obtained with sufficient accuracy by multiplying the figures found in columns 8 and 12 of Table XXXVI. by ·95.

The slide-valve faces of cylinders, when a false face is fitted, should be made of thickness given by the rule,—

**Rule 50.** Thickness of cylinder valve-face (when a false face is fitted) $= \cdot 8 \times f$,

and when no false face is used the rule becomes,—

**Rule 50a.** Thickness of cylinder valve-face (when no false face is used) $= \cdot 9 \times f$.

False faces of hard cast iron may have a thickness,—

**Rule 51.** Thickness of cast-iron false face $= \cdot 75 \times f$.

## CAST-STEEL CYLINDER COVERS.

For a flat plate secured round the edge after the manner of a cylinder cover, but yet not quite encastré, the formula may be written,—

$$t=\sqrt{\frac{3r^2p}{4f}} \quad . \quad . \quad . \quad (a)$$

where $t$ is the thickness in inches; $r$ the radius of the cylinder,—also in inches; $p$ the maximum pressure on cover in lbs. per square inch above atmosphere; and $f$ the greatest permissible stress per square inch on the material. When a suitable value of $f$ has been inserted, and a small constant quantity (necessary for practical reasons) added, this becomes,—

**Rule 52.** $$t=\sqrt{\frac{3r^2p}{160{,}000}}+\cdot25,$$

which gives the thickness (very nearly) of steel cylinder covers of the single type, ribbed as directed below, and coned to suit steel pistons, made in accordance with formulæ and Table given on pages 106, 108.

For steel covers of triple engines, working at 180 lbs. pressure, this formula (after reduction and further slight adjustment of constant) takes the forms given at heads of columns in following Table:—

### Table XXXVII.—Cast-Steel Cylinder Covers.

| Dia. of cylinder in inches. | H.P. | | | M.P. | | | L.P. | | |
|---|---|---|---|---|---|---|---|---|---|
| | Thick. in inches $\frac{D}{36}+\cdot24.$ | | No. of ribs $\frac{D+20}{6}$ | Thick. in inches $\frac{D}{56}+\cdot34.$ | | No. of ribs $\frac{D+20}{7\cdot5}$ | Thick. in inches $\frac{D}{100}+\cdot42.$ | | No. of ribs $\frac{D+20}{9}$ |
| 10 | ·52 | $\frac{9}{16}$ | 5 | ·52 | $\frac{9}{16}$ | 4 | ·52 | $\frac{9}{16}$ | 4 |
| 20 | ·79 | $\frac{13}{16}$ | 7 | ·70 | $\frac{3}{4}$ | 6 | ·62 | $\frac{5}{8}$ | 5 |
| 30 | 1·07 | $1\frac{1}{16}$ | 9 | ·87 | $\frac{7}{8}$ | 7 | ·72 | $\frac{3}{4}$ | 6 |
| 40 | 1·35 | $1\frac{3}{8}$ | 10 | 1·05 | $1\frac{1}{16}$ | 8 | ·82 | $\frac{7}{8}$ | 7 |
| 50 | 1·63 | $1\frac{5}{8}$ | 12 | 1·23 | $1\frac{1}{4}$ | 10 | ·92 | $\frac{15}{16}$ | 8 |
| 60 | 1·91 | $1\frac{15}{16}$ | 14 | 1·41 | $1\frac{7}{16}$ | 11 | 1·02 | $1\frac{1}{16}$ | 9 |
| 70 | ... | ... | ... | 1·59 | $1\frac{5}{8}$ | 12 | 1·12 | $1\frac{1}{8}$ | 10 |
| 80 | ... | ... | ... | 1·77 | $1\frac{13}{16}$ | 14 | 1·22 | $1\frac{1}{4}$ | 12 |
| 90 | ... | ... | ... | ... | ... | ... | 1·32 | $1\frac{3}{8}$ | 13 |
| 100 | ... | ... | ... | ... | ... | ... | 1·42 | $1\frac{7}{16}$ | 14 |
| 110 | ... | ... | ... | ... | ... | ... | 1·52 | $1\frac{9}{16}$ | 15 |

For higher pressures multiply the thickness given by $\sqrt{\frac{p}{180}}$, $p$ being the pressure.

The thickness of the ribs may be ·9 × thickness of cover.

The steeper cone of the H.P. cover undoubtedly gives it considerable additional strength, but its greater liability to shock and strain from the presence of water may be considered as almost balancing the account.

## CAST-IRON VALVE BOX COVERS.

Where these are circular, as in the case of covers for piston-valve boxes, a modification of the formula (*a*) given above may be again employed; the nature of the material renders the addition of a larger constant quantity necessary, and provision must be made for stiffness in L.P. covers by adding a constant to the pressure also. The formula may then be written,—

**Rule 53.** $$t = \sqrt{\frac{3 \times r^2 \times (p+15)}{4 \times 45{,}000}} + {\cdot}5.$$

This will be found to give practically the same results as Rule 44*a* and Table XXXVI., and may be used in their place for cylinder covers, if preferred.

These covers should be stiffened by radial ribs which may have a thickness of ·9 × thickness of cover, and a depth at centre of 3·5 × thickness of cover,—the outer edge of rib being a curve of the parabolic type when the cover is flat.

It will often happen that a cover can be made more or less curved, or dished one way or the other, and this should always be done where possible,—as the whole cover may then be made lighter.

The number of ribs may be the same as for cast-iron pistons (see Table XLVII.).

When weight-shaft brackets, or slide-rod guides, are attached to these covers, they must, of course, be made proportionately stronger.

For rectangular doors consisting of a single flat plate of metal of thickness ($t$), stiffened by ribs, the greatest section of which is not less than $(3 \times t) \times ({\cdot}9t)$, and which are spaced in accordance with Rule 48 (page 90), the formula is,—

**Rule 54.** $$t = \sqrt{\frac{K \times b^2 \times (p+15)}{c}} + {\cdot}5 \quad . \quad . \quad . \qquad (b)$$

where $t$ = thickness in inches.

$b$ = breadth within flanges of valve-boxes, in inches.

$$K = \frac{\text{Length}^4}{\text{Length}^4 + \text{breadth}^4}$$

$p$ = max. pressure in lbs. per sq. inch to which door is subject.

$c$ = 70,000 for single doors.

$c$ = 450,000 for double doors (see next paragraph).

When the door is rectangular and of the double type—consisting of two equal thicknesses of metal tied together and locally stiffened by ribs, in conformity with Rule 48—the over-all thickness of the doòr at centre should not be less than $\frac{\text{Shortest span}}{5}$, and the outer surface may be curved in both directions, like a "hog-back" girder. When these conditions are complied with, the formula ($b$), given above, will give the thickness of either the outer or inner metal if the value $c=450{,}000$ be employed.

Large doors of this type, when fitted to Naval engines, are commonly made in cast steel. When this material is used the over-all thickness of door at centre may be $\frac{\text{Shortest span}}{6{\cdot}5}$, and the following variant of the above formula ($b$) may be used to determine thickness of metal :—

**Rule 54a.** $$t=\sqrt{\frac{\text{K}\times b^2\times(v+15)}{250{,}000}}+{\cdot}25.$$

A convenient method of saving some space and weight, in connection with doors of this type, is to slightly increase the thickness of the inner metal, and make sufficient openings at the ends, to allow of the door itself being utilised as a passage for the steam from one end of the valve-box to the other.

## JOINTS OF CYLINDER COVERS, ETC.

As the widths and thicknesses of flanges should vary directly as the diameter of the stud employed, it is best when designing a joint to fix this diameter first.

In fixing the diameter of stud the three following conditions must be satisfied :—

(1) $\frac{\text{Total load on cover}}{\text{Effective area of 1 stud}\times\text{No. of studs}}$ must not exceed value given in Table XLIII.

**Rule 55.**—(2) The diameter of stud should not exceed ·8 × thickness of flange ; and the thickness of flange should not exceed 1·5 × thickness of metal to which it is attached.

**Rule 56.**—(3) The pitch of the studs should not be less than diameter × 3·5 (to give room for spanner).

When no water test is required a pitch of 6 diameters is often used for L.P. doors, &c.

The diameter is usually arrived at by a rough process of trial and error, as studs and bolts must be of standard size.

**Rule 57.**—The width of joint may be from diameter × 2·3 to 3, but—unless weight is of the greatest importance—it should not be less than diameter × 2·8.

The thickness of cover flange may be from 1 to 1·3 × diameter.

The limit which the pitch should in no case exceed is given by the rule,—

**Rule 58.** Maximum pitch in inches $= \sqrt{\frac{t \times 100}{p}}$

where $t$ is the thickness of cover or door flange in sixteenths of an inch, and $p$ the pressure in pounds per square inch, on the cover.

For covers of cast steel the co-efficient in the above rule may be 120 in place of 100.

It is not necessary that studs should always carry the full load allowed by Table XLI., and it is sometimes convenient to load them much more lightly when it is desired to use the same size of stud for all the covers in connection with a set of cylinders.

Where bolts are used, as in the case of a steam joint between two cylinders, the thickness of the flanges should be about diameter of bolt × 1·25, and not exceed 1·5 × (thickness of metal to which flange is attached); the width of the joint is, of course, also increased by an amount equal to about ·6 of the thickness of that metal.

## CYLINDER RELIEF OR ESCAPE VALVES.

**Rule 59.**—For the L.P. cylinder these should have a diameter of about one-fifteenth the diameter of the cylinder; for the H.P. cylinder, where priming water may have to be dealt with, the proportion may be one-eleventh; and for the M.P. cylinder, an intermediate proportion of say one-thirteenth.

In horizontal engines there should be at least one valve at each end of each cylinder—(the Admiralty requirement was two valves at each end of each cylinder)—and in vertical engines there should be one at each end of the H.P. cylinders, and one at the bottom of each of the others.

The following Table of sizes of Relief-valves is calculated to give about the above-named proportions, but the nearest half-inch is given in most cases—as it is neither usual nor necessary to make patterns for every quarter of an inch.

## Table XXXVIII.—Cylinder Relief Valves.

| Dia. of cylinder. | Dia. of Relief Valve in Ins. | | | Dia. of cylinder. | Dia. of Relief Valve in Ins. | | |
|---|---|---|---|---|---|---|---|
| | H.P. | M.P. | L.P. | | H.P. | M.P. | L.P. |
| 15 | 1½ | ... | ... | 65 | 6 | 5 | 4½ |
| 20 | 2 | 1¾ | ... | 70 | ... | 5½ | 4½ |
| 25 | 2¼ | 2 | 1¾ | 75 | ... | 6 | 5 |
| 30 | 2½ | 2¼ | 2 | 80 | ... | 6 | 5½ |
| 35 | 3 | 2½ | 2¼ | 85 | For valves over 5 ins. dia. fit two valves of equivalent area. | | 5½ |
| 40 | 3½ | 3 | 2½ | 90 | | | 6 |
| 45 | 4 | 3½ | 3 | 95 | | | 6½ |
| 50 | 4½ | 4 | 3½ | 100 | | | 6½ |
| 55 | 5 | 4½ | 3½ | 105 | | | 7 |
| 60 | 5½ | 4½ | 4 | 110 | | | 7½ |

The H.P. and L.P. columns are equally applicable to the cylinders of Compound, Triple, and Quadruple engines.

It is very desirable that all relief-valves should be fitted with guard rings, or other suitable appliances, to prevent thoughtless screwing up of the springs whenever a slight leakage occurs.

In confined engine-rooms it is also desirable to have light pipes fitted to convey any water that may escape down to the crank-pits or bilge.

For methods of calculating strengths and proportions of springs see Board of Trade Rules for Springs.

## RECEIVER SAFETY VALVES.

For small and medium sized cylinders (up to 60 inches diameter say), these may be of the same size and pattern as the cylinder relief-valves,—*i.e.*, L.P. receiver valve same as L.P. cylinder valve, and so on; but this rule makes the valves unnecessarily large for larger cylinders, and more appropriate sizes are given by the rule,—

**Rule 60.**

Diameter of Receiver Safety Valve $= \frac{\sqrt{\text{Diameter of cylinder.}}}{2}$

## DRAIN VALVES, OR COCKS, FOR CYLINDERS, RECEIVERS, ETC.

These should be of the proportions given by the following Table :—

Table XXXIX.—Drain Valves.

| Diameter of Cylinder or L.P. Cylinder, if Compound, Triple, &c. | Diameter of valve or cock. | Diameter of Cylinder or L.P. Cylinder, if Compound, Triple, &c. | Diameter of valve or cock. |
|---|---|---|---|
| Up to 10″ | ½″ | 46 to 60″ | 1¼″ |
| 11 to 15″ | ⅝″ | 61 to 75″ | 1½″ |
| 16 to 30″ | ¾″ | 76 to 90″ | 1¾″ |
| 31 to 45″ | 1″ | Above 90″ | 2″ |

In Compound, Triple, and Quadruple engines, the valves or cocks in connection with the H.P. and M.P. cylinders, should be of the same size as those for L.P. cylinder.

All pipes from H.P. and M.P. drain valves should be led to hot-well or feed tank, and should be fitted with self-acting non-return valves to prevent water getting back into cylinders. The pipes from L.P. drain valves should be led to the condenser, and also be fitted with non-return valves.

The drainage from the steam jackets should be collected in suitable vessels (each fitted with a gauge glass), and then led away to the hot-well or feed tank. The pipes should be fitted with adjustable screw-down valves, placed in sight of the gauge glasses, and within easy reach of the attendants.

Reversing engine drains should be led to the condenser, and fitted with non-return valves ; but the drainage from other auxiliary engines is better led into the bilges, as it always contains a large percentage of oil.

Each steam jacket must have its own separate adjustable drain-valve, but the cylinder and receiver drains may be, to a certain extent, grouped, or led into common pipes so as to reduce the number of valves on hot-well or feed tank, and condenser.

Drain cocks of suitable sizes should be fitted to every pipe, passage, or place where water can lodge, so that, when engines are cold, they may be quite free from water.

## STARTING, PASS, OR AUXILIARY VALVES.

In small Paddle engines it is usual to arrange the slide-valves to be worked by hand, and starting valves are therefore unnecessary.

In larger low-pressure Paddle engines, where the valves are driven by double eccentrics and links, the cuts-off are so late that starting-valves are not required.

Compound engines, with cuts-off at about six-tenths of the stroke, would occasionally be unhandy without some means of admitting steam to one of the cylinders later than the main slide allows, and starting valves should therefore be fitted to the L.P. cylinders so that steam may be admitted into either side of piston.

Triple engines, with three cranks, require only a small valve to admit steam to each receiver; the valves should be raised from their seats by means of levers acting on the spindles, and should be held shut by small spiral springs in addition to the pressure of the steam. The steam supply should be taken from the boiler side of the regulator value, and may be about one-fifth the diameter of the main steam pipe.

## COLUMN FEET AND BOLTS.

Great care should be exercised in designing these feet, for through them the load due to the steam pressure *on the cover* is transmitted; as the load is always applied suddenly, very ample section of metal should be provided to sustain it.

**Rule 61.**—The area of section through these feet should be such that the stress does not exceed 600 lbs. per square inch.

The webs from the flanges of the feet should be well spread over the cylinder bottom and towards the sides, so as to distribute the strain.

**Rule 62.** The bolts connecting the cylinder to the columns or frames should be such that the stress on them does not exceed 4000 lbs. per square inch of area at the bottom of the thread, and when there are a large number, of comparatively small size, it should not exceed 3000 lbs. per square inch.

See also Table XLIII., page 103, for loads that bolts may carry. The maximum or initial pressure should be used in calculating these stresses.

The feet should always be formed so as to permit of bolts being used (not studs), and two at least, in each foot, should be a driving fit.

## GENERAL REMARKS ON CYLINDERS.

Where weight is of great importance, as in Naval machinery, it is a common practice to shorten the cylinder, and form the port in the

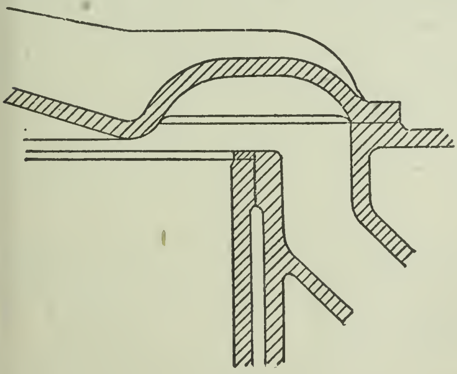


FIG. 9.

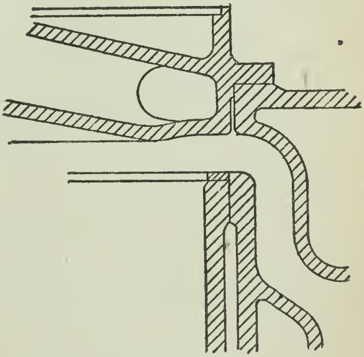


FIG. 9A.

cover, as shown in Fig. 9,—the more ordinary plan being shown by Fig. 9A.

Another method of saving weight, sometimes practised in connection with very light machinery, is to lead the exhaust steam through the relief-ring on the back of the L.P. valve, thus doing away with the exhaust port in the cylinder face, and shortening the slide valve and casing to that extent.

The corners of ports, both in false faces and in cylinder faces, should be well rounded, as the castings are very apt to crack if they are made quite square.

Steel screw-stays may be used with advantage to strengthen the various ports and passages in cylinder castings.

Where a false face is secured by the usual "cheese-headed" screws, a certain measure of relief may be given to the slide-valve, and the lubrication improved, by connecting the recesses by small grooves cut in the face.

## Horizontal Cylinders.

Horizontal cylinder barrels should always be stiffened by rings, and when the valve-box is on top—two strong ribs running up from each foot to the flange of the valve-box door, to prevent deformation of those portions of the barrel lying between the foot brackets and the valve-box, each of a thickness of ·8 × thickness of barrel, and stand out 2 × thickness of barrel. Similar ribs should also be carried round under the bottom of the cylinder from foot to foot.

In bolting horizontal cylinders down to the seatings, fitted bolts should be used at the front ends only—sufficient clearance being allowed in the other bolt holes to permit the expansion of the cylinder.

## Oscillating Cylinders.

In oscillating engines, weight is generally of great importance; the cylinder barrels are commonly made rather lighter than if the thickness were determined by the rules given (page 82). This reduction is rendered possible by the absence of any strains communicated from the framing to the cylinder, and by the great stiffness imparted to the barrel by the steam and exhaust belts, &c., and their accompanying ribs. The H.P. cylinders of compound oscillating engines should be of the thickness given by the following formula:—

**Rule 63.** Thickness of H.P. cylinder barrel $= \frac{D}{6750}(p + 50) + \cdot 45$ inch, where D is diameter of cylinder in inches, and $p$ boiler pressure.

The formula also gives the thickness of barrel for the L.P. cylinders of compound engines, and for the cylinders of simple engines working at 30 to 35 lbs. per square inch, if $p$ be taken as 25. The above thicknesses are for cylinders without liners.

The trunnions should be all of the same diameter throughout, whether engines are simple or compound, and must, of course, be large enough to allow proper area for exhaust. This plan allows of an annular air space between the steam pipe and the outer or working surface of the trunnion, and so gives a comparatively cool bearing.

The area of section where the trunnion joins the cylinder barrel should be such that the value of the expression,—

**Rule 64.**

$$\frac{\text{Max. effective pressure} \times \text{area of piston} + \text{weight of cylinder in lbs.}}{2 \times \text{area of section}}$$

does not exceed 500 ; *i.e.* the shearing stress should not exceed 500 lbs. per square inch.

A length of trunnion of about ·4 of the diameter will usually give a satisfactory amount of surface, but the area should be such that the pressure per square inch (excluding pressure due to weight of cylinder) does not exceed 350 lbs. A safe proportion, which should always be obtained where possible, is given by,—

**Rule 65.** $$\frac{\text{Max. effective pressure} \times \text{area of piston}}{2 \times \text{diameter of trunnion} \times \text{length of same}} = 300.$$

A very strong attachment of trunnion is obtained by making the outer cylinder of barrel form, and arranging the steam and exhaust belts between it and the inner cylinder or liner.

The valve faces should not be at right angles to the line joining valve-spindle and piston-rod centres, but angled so that the side next the steam entrance is nearer to the cylinder barrel than the opposite or exhaust side ; this will bring the centre of valve-spindle as close in to the cylinder as possible, whilst allowing free way for the exhaust,

The receivers of compound oscillating engines are commonly formed around the H.P. cylinder, between the inner cylinder or liner and the outer shell to which the trunnions are attached. This method has the accompanying advantage of making the outer cylinders more nearly of a size.

## CLEARANCE OF PISTON.

The following Table shows the axial clearances required in cylinders of the various diameters mentioned.

Table XL.—Piston Clearances (axial).

| Diameter of Cylinder. | Clearances. | | Diameter of Cylinder. | Clearances. | |
|---|---|---|---|---|---|
| | Crank end. | Cover end. | | Crank end. | Cover end. |
| Up to 14″ | ⅜″ | ¼″ | 61 to 80″ | ⅞″ | ⅝″ |
| 15 to 20″ | ½″ | ⅜″ | 81 to 100″ | 15/16″ | 11/16″ |
| 21 to 40″ | ⅝″ | ½″ | Above 100″ | 1″ | ¾″ |
| 41 to 60″ | ¾″ | 9/16″ | ... | ... | ... |

## STUFFING BOXES, ETC.

### Table XLI.—Stuffing-boxes for elastic packing. (Fig. 10.)

| A Diameter of Rod. | B Width of packing space. | C Diameter of Box. 1·25×A+·55 | $D_1$ 50 lbs. Depth of packing. ·9×A+·75 | $D_2$ 100 lbs. Depth of packing. 1·15×A+·75 | $D_3$ 150 lbs. Depth of packing. 1·4×A+·75 | E Overall depth of neck-ring. ·45A+·35 | F Thickn's of bushes, gland, and neck. ·045×A+·18 | Number and diameter of studs. |
|---|---|---|---|---|---|---|---|---|
| ¾″ | ⅜″ | 1½″ | 1⅜″ | 1⅝″ | 1¾″ | ¾″ | 3/16″ | 2– ⅜″ |
| ⅞ | ⅜ | 1⅝ | 1½ | 1¾ | 2 | ¾ | 3/16 | 2– ⅜ |
| 1 | 7/16 | 1⅞ | 1⅝ | 2 | 2¼ | ⅞ | ¼ | 2– ½ |
| 1⅛ | 7/16 | 2 | 1¾ | 2⅛ | 2⅜ | ⅞ | ¼ | 2– ½ |
| 1¼ | 7/16 | 2⅛ | 1⅞ | 2¼ | 2½ | ⅞ | ¼ | 2– ½ |
| 1⅜ | 7/16 | 2¼ | 2 | 2⅜ | 2⅝ | 1 | ¼ | 2– ⅝ |
| 1½ | ½ | 2½ | 2⅛ | 2½ | 2⅞ | 1 | ¼ | 2– ⅝ |
| 1¾ | ½ | 2¾ | 2¼ | 2¾ | 3¼ | 1⅛. | ¼ | 2– ¾ |
| 2 | 9/16 | 3⅛ | 2½ | 3⅛ | 3½ | 1¼ | ¼ | 2– ¾ |
| 2¼ | 9/16 | 3⅜ | 2¾ | 3½ | 3⅞ | 1⅜ | 5/16 | 2– ⅞ |
| 2½ | ⅝ | 3¾ | 3 | 3¾ | 4¼ | 1½ | 5/16 | 2– ⅞<br>3– ¾ |
| 2¾ | ⅝ | 4 | 3¼ | 4 | 4⅝ | 1⅝ | 5/16 | 2–1<br>3– ¾ |
| 3 | 11/16 | 4⅜ | 3½ | 4¼ | 5 | 1¾ | 5/16 | 2–1<br>3– ⅞ |
| 3¼ | 11/16 | 4⅝ | 3¾ | 4¾ | 5⅜ | 1⅞ | ⅜ | 2–1⅛<br>3– ⅞ |
| 3½ | ¾ | 5 | 4 | 5 | 5⅝ | 2 | ⅜ | 2–1⅛<br>3–1 |
| 3¾ | ¾ | 5¼ | 4¼ | 5¼ | 6 | 2⅛ | ⅜ | 2–1¼<br>3–1 |
| 4 | 13/16 | 5⅝ | 4½ | 5½ | 6⅜ | 2⅛ | ⅜ | 2–1¼<br>3–1 |
| 4½ | ⅞ | 6¼ | 4¾ | 6 | 7 | 2⅜ | 7/16 | 2–1⅜<br>3–1⅛ |
| 5 | 15/16 | 6⅞ | 5 | 6½ | 7¾ | 2⅝ | 7/16 | 2–1⅜<br>3–1⅛ |
| 5½ | 1 | 7½ | 5⅝ | 7 | 8½ | 2⅞ | 7/16 | 4–1⅛ |
| 6 | 1⅛ | 8¼ | 6¼ | 7⅝ | 9¼ | 3 | 7/16 | 4–1⅛ |
| 6½ | 1⅛ | 8¾ | 6⅝ | 8⅛ | 9⅞ | 3¼ | ½ | 4–1⅛ |
| 7 | 1¼ | 9½ | 7 | 8¾ | 10½ | 3½ | ½ | 4–1¼ |
| 7½ | 1¼ | 10 | 7½ | 9⅜ | 11¼ | 3¾ | ½ | 4–1¼ |
| 8 | 1⅜ | 10¾ | 8 | 10 | 12 | 4 | 9/16 | 4–1¼ |
| 8½ | 1⅜ | 11¼ | 8½ | 10½ | 12¾ | 4¼ | 9/16 | 4–1⅜ |
| 9 | 1½ | 12 | 8⅞ | 11 | 13⅜ | 4⅜ | 9/16 | 4–1⅜ |
| 9½ | 1½ | 12½ | 9⅜ | 11⅝ | 14⅛ | 4⅝ | ⅝ | 4–1⅜ |
| 10 | 1½ | 13 | 9¾ | 12¼ | 14¾ | 4⅞ | ⅝ | 4–1⅜ |

In all cases, the amount that the gland can enter (G) should be one-half the depth of the stuffing box ($D_1$, $D_2$, or $D_3$).

The width of packing space (B) given in above Table assumes that all glands are of gun-metal, as is usual in naval practice; if it is desired to use cast-iron glands bushed with gun-metal the width of space must be increased ⅛ inch in each case.

| | | | | | |
|---|---|---|---|---|---|
| A 3/16" | bush for a cast-iron | gland may be | used up to | 1¼" | dia. of rod. |
| ¼" | ,, | ,, | ,, | 2¾" | ,, |
| 5/16" | ,, | ,, | ,, | 3¾" | ,, |
| ⅜" | ,, | ,, | ,, | 6" | ,, |
| 7/16" | ,, | ,, | ,, | 7½" | ,, |
| ½" | ,, | ,, | ,, | 9" | ,, |
| 9/16" | ,, | ,, | ,, | 10½" | ,, |

Where two numbers and sizes of studs are given, the larger studs are intended for use with pinion nuts and toothed rings.

The sizes of studs for rods of 5½ inches diameter and upwards are fixed on the assumption that pinion nuts and toothed rings will always be used.

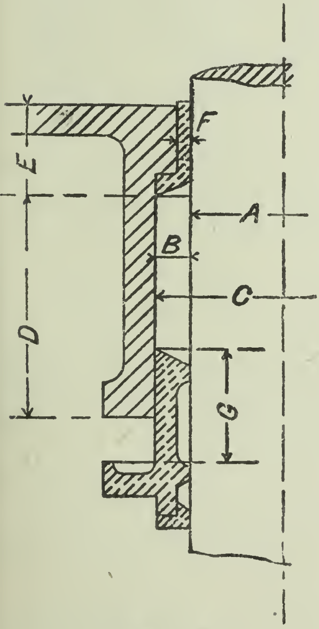


FIG. 10.

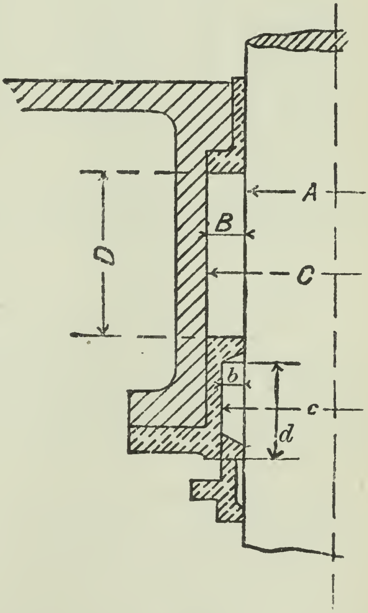


FIG. 11.

Table XLII.—Stuffing-boxes for Metallic Packing. (Fig. 11.)

| A. Diameter of rod. | B. Width of packing space. | C. Diameter of box. 1·2A+1·25 | D. Depth of metallic packing. $2\sqrt{A}+\cdot 5$. | *b.* Width of packing space. | *c.* Diameter of box. | *d.* Depth of elastic packing. $\frac{D}{3}+1\cdot 15$. |
|---|---|---|---|---|---|---|
| 3 | 1 | 5 | 4 | ¾ | 4½ | 2⅛ |
| 3¼ | 1 | 5¼ | 4⅛ | ¾ | 4¾ | 2⅛ |
| 3½ | 1 | 5½ | 4¼ | ¾ | 5 | 2¼ |
| 3¾ | 1 | 5¾ | 4⅜ | ¾ | 5¼ | 2¼ |
| 4 | 1 | 6 | 4½ | ¾ | 5½ | 2⅜ |
| 4¼ | 1 | 6¼ | 4⅝ | ¾ | 5¾ | 2½ |
| 4½ | 1⅛ | 6¾ | 4¾ | ¾ | 6 | 2⅝ |
| 5 | 1⅛ | 7¼ | 5 | ¾ | 6½ | 2¾ |
| 5½ | 1⅛ | 7¾ | 5¼ | ¾ | 7 | 2⅞ |
| 6 | 1¼ | 8½ | 5½ | ¾ | 7½ | 3 |
| 6½ | 1¼ | 9 | 5⅝ | ¾ | 8 | 3⅛ |
| 7 | 1¼ | 9½ | 5¾ | ¾ | 8½ | 3¼ |
| 7½ | 1⅜ | 10¼ | 6 | ⅞ | 9¼ | 3½ |
| 8 | 1⅜ | 10¾ | 6¼ | ⅞ | 9¾ | 3⅝ |
| 8½ | 1⅜ | 11¼ | 6⅜ | ⅞ | 10¼ | 3¾ |
| 9 | 1½ | 12 | 6½ | 1 | 11 | 3⅞ |
| 9½ | 1½ | 12½ | 6¾ | 1 | 11½ | 4 |
| 10 | 1½ | 13 | 7 | 1 | 12 | 4⅛ |

For the sake of uniformity the H.P., M.P., and L.P. stuffing-boxes of triple engines are commonly made all of the same depth.

## STRENGTH, &c., OF STUDS AND BOLTS.

As small bolts and studs are subject to severe wrenching, especially by careless hands, the nominal tensile working stress on them must be much less than with larger ones and may increase as the bolts are bigger.

The Table is based on the relation,—and the loads given should not be exceeded.

**Working stress per sq. in. = (Area at bottom of thread)** $^{\frac{4}{12}}$ **× C ;**

where C = 5000 for iron or mild steel, and 1000 for strong bronzes. For iron or steel bolts above 2 inches in diameter, and gun-metal or bronze ones above 3½ inches diameter, the moment of the twisting stress is so small, proportionately, that it may be neglected.

## Table XLIII.—Strengths of Studs and Bolts.

| Outside Dia. of stud or bolt. | Iron or Mild steel. | | Strong Bronze. | |
|---|---|---|---|---|
| | Working stress in lbs. per square inch. | Working load of 1 bolt in lbs. | Working stress in lbs. per square inch. | Working load of 1 bolt in lbs. |
| ½ | 2000 | 250 | 400 | 50 |
| ⅝ | 2500 | 500 | 500 | 100 |
| ¾ | 3000 | 900 | 600 | 180 |
| ⅞ | 3400 | 1,450 | 680 | 290 |
| 1 | 3900 | 2,150 | 780 | 430 |
| 1⅛ | 4300 | 3,000 | 860 | 600 |
| 1¼ | 4700 | 4,200 | 940 | 840 |
| 1⅜ | 5100 | 5,400 | 1020 | 1,080 |
| 1½ | 5500 | 7,100 | 1100 | 1,420 |
| 1⅝ | 5800 | 8,500 | 1160 | 1,700 |
| 1¾ | 6300 | 11,000 | 1260 | 2,200 |
| 1⅞ | 6600 | 13,100 | 1320 | 2,620 |
| 2 | 7000 | 16,100 | 1400 | 3,220 |
| 2¼ | 7000 | 20,400 | 1560 | 4,560 |
| 2½ | 7000 | 26,100 | 1730 | 6,450 |
| 2¾ | 7000 | 31,200 | 1860 | 8,300 |
| 3 | 7000 | 38,100 | 2030 | 11,000 |
| 3¼ | 7000 | 44,800 | 2170 | 13,900 |
| 3½ | 7000 | 53,000 | 2350 | 17,800 |
| 3¾ | 7000 | 60,500 | 2500 | 21,600 |
| 4 | 7000 | 70,100 | 2500 | 25,000 |
| 4¼ | 7000 | 79,500 | 2500 | 28,400 |
| 4½ | 7000 | 90,300 | 2500 | 32,200 |
| 4¾ | 7000 | 100,800 | 2500 | 36,000 |
| 5 | 7000 | 113,000 | 2500 | 40,300 |
| 5¼ | 7000 | 124,600 | 2500 | 44,500 |
| 5½ | 7000 | 138,000 | 2500 | 49,200 |

Studs and bolts may be loaded to the figures given in the above table whether the load is "dead" (as in the case of a joint), or "live" (as in the case of a connecting-rod bolt), as in the latter case mild steel will always be used, and the shearing stress due to tightening up is practically absent.

Mild steel studs and bolts should always be fitted with iron nuts, as steel ones have a tendency to "seize"; for the same reason soft bronze studs, &c., should always have iron or hard bronze nuts if possible.

Gun-metal, and the various bronzes are unsatisfactory materials for small studs and bolts, not because of any lack of tensile strength,—which is often high,—but because of their very low resistance to shearing.

When iron or steel studs are used in connection with gun-metal steam or water valves, &c., they must not be allowed to penetrate into the steam or water space, or they will rapidly corrode and come loose.

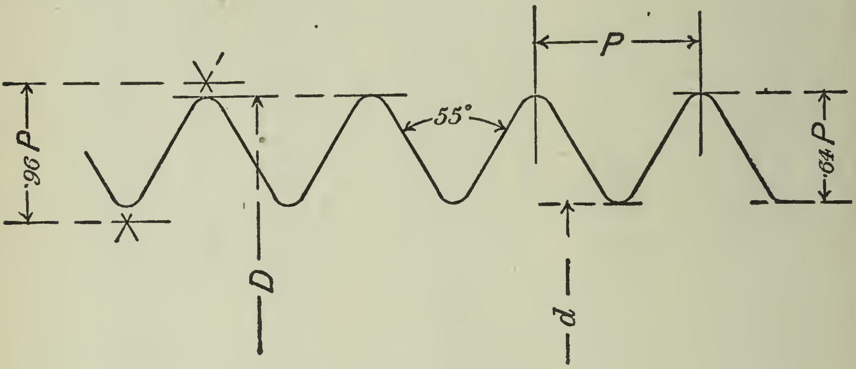
The part of a stud that is screwed into the work should be :—

**Rule 66.** Not less than 1¼ diameters long when screwed into cast-iron, and 1½ diameters when not inconvenient.

Not less than 1 diameter long when screwed into gun-metal, wrought-iron, or cast-steel.

The general dimensions, numbers of threads, &c., for bolts and nuts given in Table XLIV. are in accordance with the Whitworth Standard, and the sizes over flats and angles are the nearest sixteenth to the same.

The proportions of the Whitworth Standard thread are shown by Fig. 12.



$$d = D - 1{\cdot}28P$$

FIG. 12.

## Table XLIV.—Dimensions of Nuts, Bolts, &c.

| Dia. of Bolt. | Head and nut over flats. | Head and nut over angles. | Height of nut. | Height of head for screws and bolts. | No. of threads per inch. | Area at bottom of thread in sq. inches. | Thick. of check-nut. | Size of split-pin L. S. G. |
|---|---|---|---|---|---|---|---|---|
| ¼ | ½ | ⅝ | ¼ | 3/16 | 20 | ·027 | 3/16 | No. 14 |
| ⅜ | 11/16 | 13/16 | ⅜ | 5/16 | 16 | ·068 | ¼ | ,, 13 |
| ½ | 15/16 | 1 1/16 | ½ | 7/16 | 12 | ·121 | ⅜ | ,, 12 |
| ⅝ | 1⅛ | 1¼ | ⅝ | 9/16 | 11 | ·203 | 7/16 | ,, 11 |
| ¾ | 1 5/16 | 1½ | ¾ | 11/16 | 10 | ·303 | 9/16 | ,, 10 |
| ⅞ | 1½ | 1 11/16 | ⅞ | ¾ | 9 | ·421 | ⅝ | ,, 9 |
| 1 | 1 11/16 | 1 15/16 | 1 | ⅞ | 8 | ·554 | ¾ | ,, 8 |
| 1⅛ | 1⅞ | 2⅛ | 1⅛ | 1 | 7 | ·697 | 13/16 | ,, 7 |
| 1¼ | 2 1/16 | 2⅜ | 1¼ | 1 1/16 | 7 | ·894 | 15/16 | ,, 6 |
| 1⅜ | 2 3/16 | 2 9/16 | 1⅜ | 1 3/16 | 6 | 1·059 | 1 1/16 | ,, 5 |
| 1½ | 2 7/16 | 2 13/16 | 1½ | 1 5/16 | 6 | 1·300 | 1⅛ | ,, 4 |
| 1⅝ | 2 9/16 | 3 | 1⅝ | 1 7/16 | 5 | 1·471 | 1¼ | ,, 3 |
| 1¾ | 2¾ | 3 3/16 | 1¾ | 1 9/16 | 5 | 1·752 | 1 5/16 | ,, 2 |
| 1⅞ | 3 | 3½ | 1⅞ | 1⅝ | 4½ | 1·986 | 1⅜ | ,, 1 |
| 2 | 3⅛ | 3⅝ | 2 | 1¾ | 4½ | 2·311 | 1½ | ,, 1 |
| 2¼ | 3 9/16 | 4 1/16 | 2¼ | 2 | 4 | 2·925 | ... | 5/16 |
| 2½ | 3⅞ | 4½ | 2½ | 2 3/16 | 4 | 3·732 | ... | 5/16 |
| 2¾ | 4 3/16 | 4 13/16 | 2¾ | 2 7/16 | 3½ | 4·463 | ... | ⅜ |
| 3 | 4½ | 5¼ | 3 | 2⅝ | 3½ | 5·449 | ... | ⅜ |
| 3¼ | 4⅞ | 5⅝ | 3¼ | 2 13/16 | 3¼ | 6·406 | ... | ⅜ |
| 3½ | 5 3/16 | 6 | 3½ | 3 1/16 | 3¼ | 7·572 | ... | 7/16 |
| 3¾ | 5 9/16 | 6⅜ | 3¾ | 3¼ | 3 | 8·656 | ... | 7/16 |
| 4 | 5 15/16 | 6⅞ | 4 | 3½ | 3 | 10·026 | ... | ½ |
| 4¼ | 6⅜ | 7⅜ | 4¼ | 3¾ | 2⅞ | 11·370 | ... | ½ |
| 4½ | 6 13/16 | 7⅞ | 4½ | 3 15/16 | 2⅞ | 12·913 | ... | 9/16 |
| 4¾ | 7¼ | 8 7/16 | 4¾ | 4⅛ | 2¾ | 14·413 | ... | 9/16 |
| 5 | 7 13/16 | 9 | 5 | 4⅜ | 2¾ | 16·145 | ... | ⅝ |

The heads of all ordinary screws and bolts should be made hexagonal, and of the same size over flats as the corresponding nuts.

All check-nuts should be chamfered on both sides.

Set-screws that are frequently handled, such as those in the lock-rings on the various large nuts, should have *square* heads, and should either be of hard steel or should have their heads properly case-hardened.

Wherever a square nut is used about the engines in place of a hexagon, care should be taken that it is made of the same size as some nut over flats, as otherwise much trouble and annoyance are often caused.

## PISTONS.

For the case of a circular flat plate, supported at centre, and uniformly loaded, the formula may be written,—

$$t = C \times D\sqrt{p};$$

where $t$ = thickness, and D diameter,—both in inches; $p$ = effective pressure (or greatest difference of pressures on the two sides); and C a co-efficient.

**Cast-steel Pistons.**—These are made of a single thickness of metal, and should be coned, or "dished," to get the necessary rigidity. The three pistons of an ordinary triple engine may all be made of the same total depth, which should be such that the slope of the face of L.P. piston next crankshaft is about 1 in 3; this will give a perfectly rigid piston, and will also give room for a properly proportioned piston-rod stuffing-box.

The thickness near boss, for a piston of this type, is given very nearly by the above formula, when $C = \cdot 0046$; that is,—

**Rule 67.** $t = \cdot 0046\, D\sqrt{p}$ . . . ($a$).

For practical reasons, however, the addition of a small constant quantity, and other slight modifications, are necessary, and the formulæ for the pistons of triple engines working at about 180 lbs. pressure, and of compound engines working at about 100 lbs. pressure (since $p$ will have practically the same values in both cases) will be,—

For H.P. pistons, $t = \frac{D}{23} + \cdot 24$ (value of $p$ about 90);

For M.P. pistons, $t = \frac{D}{34} + \cdot 40$ (value of $p$ about 60);

L.P. do., $t = \frac{D}{50} + \cdot 48$ (value of $p$ about 30).

For triple engines working at 250 lbs. pressure use Rule 67, or equivalent formulæ with constant additions similar to the above, and put values of $p$ at 150, 80, and 30.

The thicknesses near the edge, or rim, should be,—

**Rule 68.** For H.P. pistons, ·65 × thickness near boss;
M.P. do., ·60 × do. do.
L.P. do., ·55 × do. do.

For the pistons of quadruple, or other engines, similar modifications of the general formula ($a$) above may be employed.*

Table XLV. is calculated by means of the above formulæ, and may be used for compound engines by simply omitting the M.P. columns.

Forged-steel pistons for the lighter machinery of Torpedo boats and Destroyers give perfectly satisfactory results, when made of thickness, near boss, given by,—

**Rule 69.** $t = \cdot 0035\ D\sqrt{p} + \cdot 2,$

where the symbols have the same values as above.

The slope of the underside of L.P. piston should not be less than 1 in 6·5. The thickness near edge should be ·6 of that near boss; and the depth of rim may be $\frac{\text{L.P. Diameter}}{12}$. They are sometimes fitted with Ramsbottom rings of cast-iron or hard bronze, and sometimes with ordinary cast-iron packing rings.

They are, as a matter of course, machined over on both sides.

Table XLVI. is calculated by means of the above formula for boiler pressures of 220 to 250 lbs.

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* *Note*—See *Trans. Inst. C.E.*, vol. cxxvii., pp. 248 *et seq.*

## Table XLV.—Cast-Steel Pistons (for 180 lbs. Working Pressure).

| Diam. of piston in inches. | H.P. Thick. in inches. |  |  |  | M.P. Thick. in inches. |  |  |  | L.P. Thick. in inches. |  |  |  |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  | Near boss. |  | Near edge. |  | Near boss. |  | Near edge. |  | Near boss. |  | Near edge. |  |
| 15 | ·89 | 15/16 | ·58 | 5/8 | ·84 | 7/8 | ·50 | 1/2 | ·78 | 13/16 | ·43 | 7/16 |
| 20 | 1·11 | 1 1/8 | ·72 | 3/4 | ·99 | 1 | ·59 | 5/8 | ·88 | 15/16 | ·48 | 1/2 |
| 25 | 1·33 | 1 3/8 | ·86 | 7/8 | 1·14 | 1 3/16 | ·68 | 11/16 | ·98 | 1 | ·54 | 9/16 |
| 30 | 1·54 | 1 9/16 | 1·00 | 1 | 1·28 | 1 5/16 | ·77 | 13/16 | 1·08 | 1 1/16 | ·59 | 5/8 |
| 35 | 1·76 | 1 3/4 | 1·14 | 1 3/16 | 1·43 | 1 7/16 | ·86 | 7/8 | 1·18 | 1 3/16 | ·65 | 11/16 |
| 40 | 1·98 | 2 | 1·29 | 1 5/16 | 1·57 | 1 9/16 | ·94 | 15/16 | 1·28 | 1 5/16 | ·70 | 3/4 |
| 45 | 2·20 | 2 1/4 | 1·43 | 1 7/16 | 1·72 | 1 3/4 | 1·03 | 1 1/16 | 1·38 | 1 3/8 | ·76 | 13/16 |
| 50 | 2·41 | 2 7/16 | 1·57 | 1 9/16 | 1·87 | 1 7/8 | 1·12 | 1 1/8 | 1·48 | 1 1/2 | ·81 | 13/16 |
| 55 | 2·63 | 2 5/8 | 1·71 | 1 3/4 | 2·02 | 2 1/16 | 1·21 | 1 1/4 | 1·58 | 1 5/8 | ·87 | 7/8 |
| 60 | 2·85 | 2 7/8 | 1·85 | 1 7/8 | 2·16 | 2 3/16 | 1·29 | 1 5/16 | 1·68 | 1 11/16 | ·92 | 15/16 |
| 65 | ... | ... | ... | ... | 2·31 | 2 5/16 | 1·38 | 1 3/8 | 1·78 | 1 13/16 | ·98 | 1 |
| 70 | ... | ... | ... | ... | 2·46 | 2 1/2 | 1·47 | 1 1/2 | 1·88 | 1 7/8 | 1·03 | 1 1/16 |
| 75 | ... | ... | ... | ... | 2·61 | 2 5/8 | 1·56 | 1 9/16 | 1·98 | 2 | 1·09 | 1 1/8 |
| 80 | ... | ... | ... | ... | 2·75 | 2 3/4 | 1·65 | 1 11/16 | 2·08 | 2 1/16 | 1·14 | 1 3/16 |
| 85 | ... | ... | .... | ... | ... | ... | ... | ... | 2·18 | 2 3/16 | 1·20 | 1 1/4 |
| 90 | ... | ... | ... | ... | ... | ... | ... | ... | 2·28 | 2 5/16 | 1·25 | 1 1/4 |
| 95 | ... | ... | ... | ... | ... | ... | ... | ... | 2·38 | 2 3/8 | 1·31 | 1 5/16 |
| 100 | ... | ... | ... | ... | ... | ... | ... | ... | 2·48 | 2 1/2 | 1·36 | 1 3/8 |
| 105 | ... | .. | ... | ... | ... | ... | ... | ... | 2·58 | 2 9/16 | 1·42 | 1 7/16 |
| 110 | ... | ... | ... | ... | ... | ... | ... | ... | 2·68 | 2 11/16 | 1 47 | 1 1/2 |

For solid cast-iron pistons of moderate size multiply edge dimensions by 1·2, and near boss by 1·7 when of good material.

Table XLVI.—Forged Steel Pistons of Torpedo Boats and Destroyers (220 to 250 lbs.).

| Diameter of piston in inches. | H.P. Thick. in inches. | | | | M.P. Thick. in inches. | | | | L.P. Thick. in inches. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Near boss. | | Near edge. | | Near boss. | | Near edge. | | Near boss. | | Near edge. | |
| 10 | ·63 | ⅝ | ·38 | ⅜ | ... | ... | ... | ... | ... | ... | ... | ... |
| 15 | ·84 | 13/16 | ·50 | ½ | ·67 | 11/16 | ·40 | ⅜ | ... | ... | ... | ... |
| 20 | 1·05 | 1 1/16 | ·63 | ⅝ | ·83 | 13/16 | ·50 | ½ | ·58 | 9/16 | ·35 | ⅜ |
| 25 | 1·27 | 1¼ | ·76 | ¾ | ·98 | 1 | ·59 | 9/16 | ·67 | 11/16 | ·40 | ⅜ |
| 30 | 1·48 | 1½ | ·89 | ⅞ | 1·14 | 1⅛ | ·68 | 11/16 | ·77 | ¾ | ·46 | 7/16 |
| 35 | ... | ... | ... | ... | 1·29 | 1¼ | ·77 | ¾ | ·87 | ⅞ | ·52 | ½ |
| 40 | ... | ... | ... | ... | 1·45 | 1 7/16 | ·87 | ⅞ | ·96 | 15/16 | ·58 | 9/16 |
| 45 | ... | ... | ... | ... | ... | ... | ... | ... | 1·06 | 1 1/16 | ·64 | ⅝ |
| 50 | ... | ... | ... | ... | ... | ... | ... | ... | 1·15 | 1⅛ | ·69 | 11/16 |
| 55 | ... | ... | ... | ... | ... | ... | ... | ... | 1·25 | 1¼ | ·75 | ¾ |

**Cast-iron Pistons.**—These must be made of thoroughly sound and good material,—at least twice melted.

A convenient unit on which to base the proportions of cast-iron box (or hollow) pistons is the thickness requisite, in a circular plate of cast-iron, to carry safely a given pressure per square inch when supported at centre and loaded uniformly.

Using again the general formula given above for such a case, viz.:—

$$t = C \times D\sqrt{p}$$

and inserting a value of $c$ that will limit the stress on the material to 3000 lbs. per square inch it becomes,—

$$t = {\cdot}008\ D\sqrt{p}$$

If the necessary constant quantity be then added, the units ($x$) for pistons of triple engines working at about 160 lbs., and of compound engines working at about 100 lbs., will be given by,—

**Rule 70.** For H.P. pistons, $x = {\cdot}008\ D\sqrt{p} + {\cdot}3$

M.P. ,, $x = {\cdot}008\ D\sqrt{p} + {\cdot}45$

L.P. ,, $x = {\cdot}008\ D\sqrt{p} + {\cdot}6$

Other proportions are then as follows:—

Rule 71. Depths at centre, ... ... ... $3{\cdot}2 \times x$.

,, 71a. Thickness of metal of faces $\begin{cases} \text{near boss} & {\cdot}33 \times (x + {\cdot}5). \\ \text{near edge} & {\cdot}3 \times (x + {\cdot}5). \end{cases}$

,, 71b. Thickness of webs, ... .. ... ${\cdot}32 \times (x + {\cdot}5)$.

,, 71c. Thickness of boss round rod, ... ${\cdot}7 \times x$.

,, 71d. Depth of packing-ring, ... ... ${\cdot}75 \times (\text{L.P.}x + 1{\cdot}5)$.

,, 71e. Thickness do. ... ... ${\cdot}22 \times (\text{L.P.}x + 1{\cdot}5)$.

,, 71f. Diameter of junk-ring bolts, ... $({\cdot}2 \times \text{L.P.}x) + {\cdot}5$.

## Table XLVII.—Proportions

| Diameter of piston in inches. | Values of $x$. $= \cdot 008 D \sqrt{p} + \begin{cases} \cdot 3 \text{ H.P.} \\ \cdot 45 \text{ M.P.} \\ \cdot 6 \text{ L.P.} \end{cases}$ | | | Depths at centre. $= 3 \cdot 2 \times x$. | | | Thickness of Metal $= \cdot 33 \times (x + \cdot 5)$ at centre and | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | H.P. | | M.P. | |
| | H.P. | M.P. | L.P. | H.P. | M.P. | L.P. | Centre. | Edge. | Centre. | Edge. |
| 15 | 1·44 | 1·37 | 1·25 | 4·60 | 4·38 | 4·00 | ·64 | ·58 | ·62 | ·56 |
| 20 | 1·82 | 1·68 | 1·46 | 5·82 | 5·37 | 4·67 | ·76 | ·69 | ·72 | ·65 |
| 25 | 2·20 | 1·99 | 1·68 | 7·04 | 6·37 | 5·36 | ·89 | ·81 | ·82 | ·75 |
| 30 | 2·58 | 2·30 | 1·89 | 8·25 | 7·35 | 6·05 | 1·01 | ·92 | ·92 | ·84 |
| 35 | 2·96 | 2·61 | 2·11 | 9·47 | 8·33 | 6·75 | 1·14 | 1·04 | 1·02 | ·93 |
| 40 | 3·34 | 2·91 | 2·33 | 10·69 | 9·31 | 7·45 | 1·27 | 1·15 | 1·12 | 1·02 |
| 45 | 3·72 | 3·22 | 2·55 | 11·91 | 10·30 | 8·14 | 1·40 | 1·27 | 1·23 | 1·11 |
| 50 | 4·10 | 3·53 | 2·76 | 13·12 | 11·29 | 8·83 | 1·52 | 1·38 | 1·33 | 1·20 |
| 55 | 4·48 | 3·84 | 2·98 | 14·34 | 12·27 | 9·52 | 1·65 | 1·49 | 1·43 | 1·30 |
| 60 | 4·86 | 4·14 | 3·19 | 15·55 | 13·25 | 10·21 | 1·77 | 1·60 | 1·53 | 1·39 |
| 65 | ... | 4·45 | 3·41 | ... | 14·24 | 10·90 | ... | ... | 1·63 | 1·48 |
| 70 | ... | 4·76 | 3·62 | ... | 15·23 | 11·58 | ... | ... | 1·73 | 1·57 |
| 75 | ... | 5·07 | 3·84 | ... | 16·22 | 12·27 | ... | ... | 1·84 | 1·67 |
| 80 | ... | 5·38 | 4·05 | ... | 17·21 | 12·96 | ... | ... | 1·94 | 1·76 |
| 85 | ... | ... | 4·27 | ... | ... | 13·66 | ... | ... | ... | ... |
| 90 | ... | ... | 4·49 | ... | ... | 14·37 | ... | ... | ... | ... |
| 95 | ... | ... | 4·71 | ... | ... | 15·06 | ... | ... | ... | ... |
| 100 | ... | ... | 4·92 | ... | ... | 15·74 | ... | ... | ... | ... |
| 105 | ... | ... | 5·14 | ... | ... | 16·43 | ... | ... | ... | ... |
| 110 | ... | ... | 5·35 | ... | ... | 17·12 | ... | ... | ... | ... |

of Cast-Iron Pistons

| of Faces. ·3×($x$+·5) at edge. | | Thickness of boss round rod ·7×$x$. | | Number of Webs. | | | Packing Ring. | | Dia. of Junk-ring bolts (L.P.$x$×·2)+·5 |
|---|---|---|---|---|---|---|---|---|---|
| L.P. | | | | H.P. | M.P. | L.P. | Depth ·75 × (L.P.$x$+1·5). | Thick. ·22 × (L.P.$x$+1·5). | |
| Centre. | Edge. | H.P. | L.P. | $\frac{D+20}{9}$ | $\frac{D+20}{10}$ | $\frac{D+20}{11}$ | | | |
| ·58 | ·52 | 1·01 | ·87 | 4 | ... | ... | 2·06 | ·60 | ·75 |
| ·65 | ·59 | 1·27 | 1·02 | 5 | 4 | 4 | 2·22 | ·65 | ·79 |
| ·72 | ·66 | 1·54 | 1·17 | 5 | 5 | 5 | 2·38 | ·70 | ·84 |
| ·79 | ·72 | 1·81 | 1·32 | 6 | 5 | 5 | 2·54 | ·75 | ·88 |
| ·86 | ·79 | 2·07 | 1·48 | 6 | 6 | 5 | 2·71 | ·80 | ·92 |
| ·93 | ·85 | 2·34 | 1·63 | 7 | 6 | 6 | 2·87 | ·84 | ·96 |
| 1·01 | ·92 | 2·60 | 1·78 | 8 | 7 | 6 | 3·03 | ·89 | 1·01 |
| 1·08 | ·98 | 2·87 | 1·93 | 8 | 7 | 7 | 3·19 | ·94 | 1·05 |
| 1·15 | 1·05 | 3·13 | 2·08 | 9 | 8 | 7 | 3·36 | ·99 | 1·10 |
| 1·22 | 1·11 | 3·40 | 2·23 | 9 | 8 | 8 | 3·52 | 1·03 | 1·14 |
| 1·30 | 1·17 | ... | 2·38 | ... | 9 | 8 | 3·68 | 1·08 | 1·18 |
| 1·37 | 1·23 | ... | 2·53 | ... | 9 | 9 | 3·84 | 1·13 | 1·22 |
| 1·44 | 1·30 | ... | 2·68 | ... | 10 | 9 | 4·00 | 1·18 | 1·27 |
| 1·50 | 1·36 | ... | 2·83 | ... | 10 | 9 | 4·16 | 1·22 | 1·31 |
| 1·58 | 1·43 | ... | 2·99 | ... | ... | 10 | 4·33 | 1·27 | 1·36 |
| 1·66 | 1·50 | ... | 3·14 | ... | ... | 10 | 4·49 | 1·32 | 1·40 |
| 1·73 | 1·56 | ... | 3·29 | ... | ... | 11 | 4·65 | 1·37 | 1·44 |
| 1·80 | 1·62 | ... | 3·44 | ... | ... | 11 | 4·81 | 1·41 | 1·48 |
| 1·87 | 1·69 | ... | 3·59 | ... | ... | 12 | 4·97 | 1·46 | 1·53 |
| 1·93 | 1·75 | ... | 3·74 | ... | ... | 12 | 5·13 | 1·51 | 1·57 |

| | | |
|---|---|---|
| **Rule 72.** | Pitch of junk-ring bolts, | H.P.—7½ diameters.<br>M.P.—8½ do.<br>L.P.—10 do. |
| **Rule 73.** | Spring pressures per square inch of packing-ring in contact with cylinder,... ... ... ... | H.P.—3½ lbs.<br>M.P.—3 lbs.<br>L. P.—2½ lbs. |
| **Rule 74.** | Number of webs in piston, ... | H.P.—$\frac{D+20}{9}$.<br>M.P.—$\frac{D+20}{10}$.<br>L.P.—$\frac{D+20}{11}$. |

This spacing of webs will ensure that Rule 48, page 90, is not violated. In calculating *local* strength of piston face, care must be taken to use the gauge pressure acting on it, and not the effective pressure on piston as in above rules. A good method of increasing local strength, and at the same time assisting the moulder, is to cast a solid bar as a tie at the centre of gravity of each sector of the piston : the diameter of these ties may be $\cdot 6 \times$ L.P.$x$ All holes provided for the removal of core material should be strengthened by an internal ring, unless through the circular rim.

Pistons of all Compound engines are usually, for practical reasons, made all of the same depth (except in the case of tandem engines) ; but, if a piston is made of less depth than that given in Table XLVII., the metal of the faces must be increased in proportion.

The proportions given for packing-rings and junk-ring bolts are equally applicable to cast-steel pistons.

All junk-ring bolts (or nuts, if studs are used) should be locked by means of a light wrought-iron ring, secured to junk-ring by studs having square bodies, which stand up through guard-ring, and through which stout split-pins are fitted : ⅝-inch studs and No. 1 L.S.G. split-pins are commonly used for large pistons.

When the ordinary packing-ring is used an excellent piston is produced by placing behind it,—one between every pair of junk-ring bolts,—a number of short spiral springs acting radially, and adjusted to give the pressures named above. For the pistons of auxiliary engines the Cameron spring (a corrugated steel ribbon) is most useful, and is also often employed for L.P. pistons of large size. The old-fashioned coach spring should never be employed.

In horizontal engines solid packings should be used in lieu of springs for the lower one-third circumference of piston ; diagonal and oscillating pistons are also better for some solid packings.

Table XLVII. is calculated by means of the above formulæ, and shows at a glance the various proportions of H.P., M.P., & L.P. pistons, from 15 inches to 110 inches diameter, for Triple engines working at about 160 lbs., and Compound engines working at about 100 lbs. pressure.

When using Table XLVII. for Compound engines simply omit M.P. columns. The dimensions are given in inches and decimals only, as the proportions for pistons of intermediate diameter are more readily seen with this notation.

## PISTON-RODS.

These should be made of steel of 38 tons tensile; softer steels do not wear satisfactorily. (**Rule 75**) the maximum permissible stress per square inch at bottom of thread should not exceed 10,500 for high tensile steel, and for wrought-iron and mild steel 8400.

For small rods it is not advisable, in ordinary work, to exceed the limits given in the following rule :—

**Working stress per sq. in.** $= (\textbf{Area at bottom of thread})^{\frac{1}{3}} \times \mathbf{C}$,
where $\mathbf{C} = 5000$ for strong steel and 3800 for iron and mild steel.

### Table XLVIII.—Strengths of Piston-Rods.

| Diameter of piston-rod screw in inches. | 38-Ton Steel. | | Iron and Mild Steel. | |
|---|---|---|---|---|
| | Working stress in lbs. per sq. inch. | Effective strength of rod in lbs. | Working stress in lbs. per sq. inch. | Effective strength of rod in lbs. |
| ⅞ | 3,700 | 1,560 | 2800 | 1,180 |
| 1 | 4,100 | 2,270 | 3100 | 1,720 |
| 1⅛ | 4,400 | 3,070 | 3300 | 2,270 |
| 1¼ | 4,800 | 4,290 | 3600 | 3,220 |
| 1⅜ | 5,100 | 5,400 | 3900 | 4,130 |
| 1½ | 5,400 | 7,020 | 4100 | 5,330 |
| 1⅝ | 5,700 | 8,380 | 4300 | 6,320 |
| 1¾ | 6,000 | 10.500 | 4600 | 8,060 |
| 1⅞ | 6,300 | 12,500 | 4800 | 9,530 |
| 2 | 6,600 | 15,200 | 5000 | 11,500 |
| 2¼ | 7,000 | 20,400 | 5300 | 15,500 |
| 2½ | 7,400 | 27,600 | 5600 | 20,900 |
| 2¾ | 7,800 | 34,800 | 5900 | 26,300 |
| 3 | 8,200 | 44,700 | 6100 | 33,200 |
| 3¼ | 8,600 | 55,000 | 6400 | 41,000 |
| 3½ | 9,000 | 68,000 | 6800 | 51,500 |
| 3¾ | 9,250 | 80,000 | 7200 | 62,300 |
| 4 | 9,500 | 95,200 | 7600 | 76,100 |
| 4½ | 10,000 | 130,000 | 8000 | 103,300 |
| 5 &c. | 10,500 | ... | 8400 | ... |

A convenient form of the formula for the body of rod is then,—

**Rule 76.** **Diameter of piston-rod** $= \dfrac{\text{Diameter of cylinder}}{\text{F}} \times \sqrt{p},$

where $p$ = greatest effective pressure on the piston; F a co-efficient whose values are,—

| | |
|---|---|
| Naval engines,—direct-acting, | 50 |
| Mercantile,—ordinary stroke, direct-acting, | 45 |
| ,, long ,, ,, | 42 |
| ,, very long ,, ,, | 41 |
| ,, medium stroke, oscillating, | 40 |
| ,, long ,, ,, | 38 |

The stroke of Naval engines rarely exceeds dia. of L.P. cylinder × ·6 and is usually between ·5 and ·6 × L.P. diameter.

"Ordinary" stroke, for the engines of a merchant steamer, is,—diameter of L.P. cylinder × ·60 to ·66; whilst ·7 to ·8 would be "long," and ·8 to 1 "very long."

A "medium" stroke for an oscillating engine is,—diameter of cylinder × ·95; or, if a compound engine,—diameter of L.P. cylinder × ·8.

Compound diagonal engines would come under the heading "long," since they are usually given a stroke of L.P. diameter × ·75, though the practice of different makers varies between ·7 and 1·3 × L.P. diameter.

**Rule 76a. Area of section of piston-rod** $= \dfrac{\text{Area of cylinder} \times p}{F^2}$

where $F^2$ will then be the stress in lbs. per square inch,—in each case the square of the numerical co-efficient given above, viz.:—

| | |
|---|---|
| Naval engines, direct-acting, | 2500 |
| Mercantile, ordinary stroke, direct-acting, | 2000 |
| ,, long ,, ,, | 1800 |
| ,, very long ,, ,, | 1700 |
| ,, medium stroke, oscillating, | 1600 |
| ,, long ,, ,, | 1450 |

**Attachment of Piston to Rod.**—In small auxiliary engines it is a common practice simply to reduce the diameter of the piston-rod about ¼ inch, and fit the piston up against the shoulder; but above 15 inches diameter of piston it is better to leave only as much shoulder as may be desired for "trueing-up" the rod at some future time, and then give a taper of 1 in 4 on the diameters (**Rule 77**) (*i.e.* if taper is 4 inches long, the smaller diameter will be 1 inch less than the larger), and continue it until it dies away into the parallel part near screw, the diameter of which has been determined as directed above.

Where a piston-rod is fitted into a crosshead, the same shoulder, taper, &c., should be used as for a piston.

The depth of piston-rod nut need not exceed the diameter of the screw; and it may be recessed as far into the piston as desired, as the strength of the piston is not materially effected thereby.

The piston-rods of oscillating engines are commonly made with a sort of cylindrical "bolt-head," which is recessed into the lower face of the piston, whilst the nut is similarly recessed (to its full depth) into the upper face.

The cast-steel pistons of large Naval engines are sometimes made without any central boss, and attached to a circular flange (resembling a shaft coupling) on the piston-rod by means of a number of comparatively small bolts. This method, which was first used by the late Dr Kirk, has many good points, but the chief is perhaps the ease with which the connection can be made and broken again.

**Rule 78.** The diameter of this flange is about **diameter of piston-rod × 2·25**, and its thickness about **diameter of piston-rod × ·33**. The attaching bolts (usually between 2½ inches and 3½ inches diameter) should be of mild steel, but should not carry more than 6000 lbs. per square inch.

## PISTON-ROD GUIDES AND GUIDE BLOCKS.

Piston-rod guides should, wherever possible, be made of hard close-grained cast-iron, and the guide blocks, or crosshead shoes, should be made of the same material, since no combination of metals gives better results when in good working condition than cast-iron rubbing on cast-iron.

White metal is often used for the faces of the shoes (either fitted or cast into recesses in the cast-iron or gun-metal shoes), and gives results little or no better than those obtained with cast-iron on cast-iron, when the surfaces, etc., are *pari passu*, but with white metal there is no danger of abrasion, or from sparking in an enclosed engine.

Let $r$ be the radius of crank and $l$ the length of the connecting-rod; $\alpha$ the angle made by the rod with the axial line of cylinder. When P is the effective pressure on the piston; Q the pressure on guide;

$$\text{Then } Q = P \tan \alpha.$$

When the cut-off is later than half stroke, the maximum pressure will be when the crank is at right angles to the axial line.

$$\text{That is } Q = P \frac{r}{\sqrt{l^2 - r^2}},$$

as the connecting-rods in marine engines are generally twice the stroke, and $l = 4r$.

$$\text{Then, maximum pressure on guide } Q = \frac{P}{3 \cdot 87}.$$

The common method of fixing the sizes of crosshead shoes is to assume that the pistons, when at half stroke, have still the maximum effective pressures acting on them; to calculate the pressures on the guides from these loads in the manner described above; and then to fix the areas of the shoes so that the pressure per square inch may not exceed 80 lbs. (**Rule 79.**) Pressure per square inch $= (1760 \div \sqrt{S + 100})$ lbs., S being the speed of piston in feet per minute. Of course, where the indicator diagrams from similar engines are available, the actual effective pressures at half stroke may be ascertained and used, but the pressures per square inch may then be more than given by Rule 79. It may also be 30 to 50 per cent. more with efficient forced lubrication.

It is very desirable to keep the crossheads and guides of compound and triple engines all of the same dimensions, and to effect this, the *mean* of the piston loads may be used to calculate from. The surfaces of the astern shoes may be from ·5 to ·7 of those of the ahead ones.

For gudgeons, and bolts and caps, see under **Connecting-Rods.**

Where the gudgeon brasses are recessed into the piston-rod head, the inner or half-round brass may have a thickness,—

**Rule 80.** **Thickness of inner half brass** $= \dfrac{\text{Diam. of gudgeon}}{8} + \cdot 2,$

whilst for the outer or flat half the thickness may be,—

**Rule 80a.** **Thickness of outer half brass** $= \dfrac{\text{Diam. of gudgeon}}{7} + \cdot 2.$

See note with reference to white metal under "Connecting-rod gudgeons," page 118, also Table XLIX., page 121.

## CONNECTING-RODS.

The length of the connecting-rod, measured from centre of gudgeon to centre of crank pin, should not be less than twice the stroke of the piston.

So far as tensile stress alone is concerned, a rod of the same diameter as the piston-rod at the bottom of the thread would suffice, but there are also to be considered compressive stress, bending stress due to the inertia of the rod itself, and bending stress due to the friction of the gudgeon and crank pin.

As marine connecting-rods are usually between 10 and 16 diameters (measured at mid-length) long, they must, when compressive stresses are under consideration, be treated as struts jointed at the ends.

Hodgkinson & Gordon's formula for the *breaking* strength of such a strut, of circular section, is,—

$$P = \frac{fs}{1 + 4a\dfrac{l^2}{d^2}}$$

where P is breaking load in lbs.; $l$ length from centre to centre, and $d$ diameter, both in inches; $s$ area of section in square inches; and $f$ and $a$ co-efficients, whose values for wrought-iron or mild steel are 36,000 and $\frac{1}{3000}$ respectively.

It should be noted that the ratio of greatest thrust on the connecting-rod is to the effective load on the piston is 1·034.

As the ratio of length to diameter rarely exceeds 16, however, it is probable that the following empirical rule (suggested by Grashof) will give more accurate results,—

**Rule 81.**
$$P=\frac{KAI}{\frac{Al^2}{c}+I}$$

where P is greatest safe load in lbs.; A sectional area of rod in sq. ins.; I moment of inertia (equal to $\cdot 05d^4$ for circular section); $l$ length in inches from centre to centre; and K and $c$ constants whose values are respectively 12,000 and 5000 for steel, and 10,000 and 5600 for wrought-iron. The maximum working load on a connecting-rod should not exceed ·75P, and is commonly about ·6P.

The bending stress, due to the inertia, may be found from the formula,—

**Rule 82.**
$$f=\cdot 137\,\frac{v^2l^2}{gRd}$$

where $v$ is velocity of crank-pin in feet per second; $l$ length of rod from centre to centre, and $d$ diameter of rod, both in inches: R radius of crank in feet; and $g=32$.

The value of $f$ will be found to lie between 800 and 1800 lbs. per square inch for the various types of engine and numbers of revolutions met with in ordinary practice, and this value must be added to the compressive stress in estimating the total stress on the material of the rod.

In a three-crank engine, when one of the crank-pins heats, the bending stress set up in the connecting-rod by the continued action of all the pistons may be very great (if a "seize" could really occur the rod would be destroyed instantly), but its magnitude cannot be even approximately calculated, and the case is, therefore, one that can only be met by an empirical formula. The above formulæ may, therefore, be used as *checks*, but the following empirical rule will be more readily applied, and the results given by it will be found to agree very closely with good modern practice:—

**Rule 83.**
$$D=\frac{\sqrt{L\times K}}{4}$$

where D is diameter of rod at middle, and L length from centre to centre, both in inches; and

$K=\cdot 028\sqrt{\text{Effective load on piston in lbs.}}$—for Mercantile engines.

$K=\cdot 022\sqrt{\text{Effective load on piston in lbs.}}$—for Naval engines.

Connecting-rods are usually made tapered from the gudgeon end to the middle, and parallel from the middle to the crank end.

**Rule 84.**—D is the diameter of piston-rod by rule;
Diameter of gudgeon fixed in rod $= 1\cdot125 \times D$;
Diameter of each gudgeon in crosshead $= 0\cdot75 \times D$;
The length of gudgeon such that:—

**Rule 85.**—The loads on gudgeon bearings, from maximum effective pressures on pistons, should never exceed $(12{,}500 \div \sqrt{R+100})$ lbs. per square inch, R being the revolutions per minute.

**Rule 85a.**—The diameter of the gudgeon (when shrunk into the connecting-rod) may be $1\cdot25 \times$ Diameter of piston-rod, and the length is then given by,—

$$\textbf{Length of Gudgeon} = \frac{\text{Max. effective load on piston} \times \sqrt{R+100}}{\text{Diameter} \times 12{,}500}.$$

**Connecting-Rod Gudgeons.**—The rods of small engines may have the gudgeon shrunk into the jaws or sides of the double-eye, and working in brasses fitted into a recess in the piston-rod head, and secured by a cap and bolts; but larger engines should have the gudgeon shrunk into or formed with the piston-rod head or crosshead, and both jaws of the connecting-rod fitted with brasses, caps, and bolts, &c. The great advantages of the latter plan are,—the ease with which the brasses can be overhauled and adjusted, and the fact that it does not require the piston-rod crosshead to be forged solid with the rod. It also braces or "trusses" the joint between the two rods, and, in a measure, compensates for the absence of side guides, which experience has shown are better omitted; and, further, should unequal wear of the brasses take place, it occurs at a distance from the axis, and is therefore of less relative importance.

Where gudgeons cannot conveniently be made of hard steel (say 40 tons tensile), they must be case-hardened, and then carefully ground true. Simple mild steel is not suitable for gudgeons or other similar parts on which pressure is great, angular movement small, and direction of motion frequently reversed; and, similarly, hard gun-metal makes a better gudgeon bearing than any ordinary white metal.

**Connecting-Rod Caps and Bolts.**—The stresses on bolts of piston-rods and connecting-rods, per square inch area at bottom of thread, may be those given in Table XLIII., page 103, the bolts being in all cases of mild steel; but, where the load is carried by four bolts in place of two, the stress per square inch should be one-eighth less,—to allow for possible inequality in screwing up.

The bodies of the bolts should be turned down to the same diameter as the bottom of the thread, except where necessary for steadying the caps and brasses, and these remaining plain portions should be of slightly larger diameter than the screw thread,—say $\frac{1}{32}$ inch up to $2\frac{1}{2}$ inch diameter, and $\frac{1}{16}$ inch for larger sizes.

The nuts should be of wrought-iron or hard steel—as mild steel nuts on steel bolts have a great tendency to "seize" and tear unless well greased.

The necessary section of cap is most readily determined by means of the ordinary formula for a beam supported at the ends, and uniformly loaded, viz. :—

**Rule 86.**
$$w=\frac{8fz}{l^2}$$

where $w$ is the working load per inch of length for any given value of $f$; $l$ is the length in inches, from centre of bolt to centre of bolt; and $z$ the modulus of the section,—equal to $\frac{\text{Breadth}\times\text{Depth}^2}{6}$ for a rectangular section. For wrought-iron, or mild steel, the values of $f$ may be,—

Flat-backed brasses and caps,—load carried by one cap,—$f=9{,}000$.
Half-round ,, ,, ,, $f=10{,}000$.
Flat-backed ,, ,, two caps,—$f=8{,}000$.
Half-round ,, ,, ,, $f=9{,}000$.

In estimating the breadths of caps, care should be taken to deduct the diameters, or breadths, of all oil holes.

The caps of connecting-rods of large mercantile engines are often made of tough cast-iron or cast-steel lined with white metal instead of half a loose shell and an ordinary cap. The shells of large rods are generally of tough cast-iron, which is really better than bronze for the carrier of white metal.

**Connecting-Rod Brasses.**—The over-all thickness of these (*i.e.* the thickness including any white metal) should be as follows :—

**Rule 87.** Flat-backed brasses for gudgeon end,—
$$\text{Thickness}=\frac{D}{7}+{\cdot}2.$$

**Rule 87a.** Flat-backed brasses for crank end,—
$$\text{Thickness}=\frac{D}{8}+{\cdot}2.$$

**Rule 87b.** Round brasses for gudgeon end,—
$$\text{Thickness}=\frac{D}{8}+{\cdot}2.$$

**Rule 87c.** Round brasses for crank end,—
$$\text{Thickness}=\frac{D}{9}+{\cdot}2.$$

**Rule 87d.**—Thickness of white metal $={\cdot}02D+{\cdot}125$.

**Rule 87e.**—When no white metal is used at crank end the thicknesses may be $\begin{cases} \frac{D}{10} + {\cdot}15 \text{ when flat backed.} \\ \frac{D}{11} + {\cdot}12 \quad \text{,, round ,,} \end{cases}$

See Table XLIX., page 121, for proportions given by these rules.

**Rule 88.**—The width of the crank end of the connecting-rod should not be less than ·7 × length of crank-pin,—as otherwise the brasses will not be properly supported.

The proportions of the double-eye at gudgeon end (fig. 13) should be as follows:—

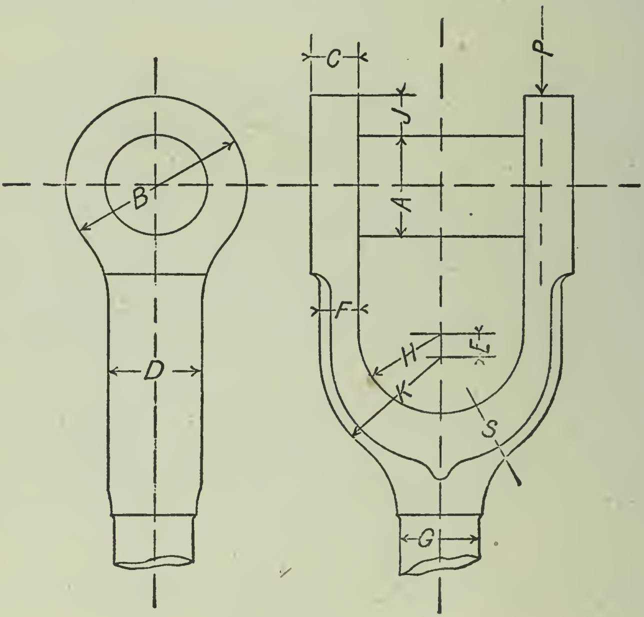


Fig. 13.

**Rules 89 and 89a.** $\frac{B}{A} = 1\ 82$, and $\frac{C}{A} = {\cdot}475$.

**Rules 89b and 89c.** $\frac{D}{G} = 1{\cdot}2$, and $\frac{F}{G} = {\cdot}45$.

The sectional area at D, or F, is then about ·62 of that at G; and area C × J about ·39 of that at G.

**Rule 89d.** Also $\frac{E}{H} = {\cdot}275$.

## Table XLIX.—Thickness of Brasses.

| Diameter of gudgeon or pin. | Description of Bearing or Brasses. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Connecting rod gudgeon when in piston rod head. | | Connecting rod gudgeon when on forked end of connect. rod. | | Crank-pins. | | | Main Bearings. | | |
| | Inner half. $\frac{D}{8}+\cdot2$ | Outer half. $\frac{D}{7}+\cdot2$ | Round brass's. $\frac{D}{8}+\cdot2$ | Flat backed bras's. $\frac{D}{7}+\cdot2$ | Round brasses. $\frac{D}{9}+\cdot2$ | Flat backed brasses. $\frac{D}{8}+\cdot2$ | White metal. $\cdot02D+\cdot125$ | Round brasses. $\frac{D}{10}+\cdot25$ | Flat backed brasses. $\frac{D}{9}+\cdot3$ | White metal. $\cdot02D+\cdot125$ |
| 2 | ·45 | ·48 | ·45 | ·48 | $\frac{D}{10}+\cdot12$ ·32 | $\frac{D}{9}+\cdot15$ ·37 | ... | $\frac{D}{11}+\cdot12$ ·30 | $\frac{D}{10}+\cdot15$ ·35 | ... |
| 2½ | ·51 | ·56 | ·51 | ·56 | ·37 | ·43 | ... | ·34 | ·40 | ... |
| 3 | ·57 | ·63 | ·57 | ·63 | ·42 | ·48 | ... | ·39 | ·45 | ... |
| 3½ | ·64 | ·70 | ·64 | ·70 | ·47 | ·54 | ... | ·45 | ·50 | ... |
| 4 | ·70 | ·77 | ·70 | ·77 | ·64 | ·70 | ·20 | ·65 | ·74 | ·20 |
| 4½ | ·76 | ·84 | ·76 | ·84 | ·70 | ·76 | ·21 | ·70 | ·80 | ·21 |
| 5 | ·82 | ·91 | ·82 | ·91 | ·75 | ·82 | ·22 | ·75 | ·85 | ·22 |
| 5½ | ·89 | ·99 | ·89 | ·99 | ·81 | ·89 | ·23 | ·80 | ·91 | ·23 |
| 6 | ·95 | 1·06 | ·95 | 1·06 | ·87 | ·95 | ·24 | ·85 | ·97 | ·24 |
| 6½ | 1·01 | 1·13 | 1·01 | 1·13 | ·93 | 1·01 | ·25 | ·90 | 1·03 | ·25 |
| 7 | 1·07 | 1·20 | 1·07 | 1·20 | ·98 | 1·07 | ·26 | ·95 | 1·08 | ·26 |
| 7½ | 1·14 | 1·27 | 1·14 | 1·27 | 1·04 | 1·14 | ·27 | 1·00 | 1·14 | ·27 |
| 8 | 1·20 | 1·34 | 1·20 | 1·34 | 1·09 | 1·20 | ·28 | 1·05 | 1·19 | ·28 |
| 8½ | ... | ... | 1·26 | 1·41 | 1·15 | 1·26 | ·29 | 1·10 | 1·25 | ·29 |
| 9 | ... | ... | 1·32 | 1·48 | 1·20 | 1·32 | ·30 | 1·15 | 1·30 | ·30 |
| 9½ | ... | ... | 1·39 | 1·56 | 1·26 | 1·39 | ·31 | 1·20 | 1·36 | ·31 |
| 10 | ... | ... | 1·45 | 1·63 | 1·31 | 1·45 | ·32 | 1·25 | 1·41 | ·32 |
| 10½ | ... | .. | 1·51 | 1·70 | 1·37 | 1·51 | ·33 | 1·30 | 1·47 | ·33 |
| 11 | ... | ... | 1·57 | 1·77 | 1·42 | 1·57 | ·34 | 1·35 | 1·52 | ·34 |
| 12 | ... | ... | 1·70 | 1·91 | 1·53 | 1·70 | ·36 | 1·45 | 1·63 | ·36 |
| 13 | ... | ... | 1·82 | 2·06 | 1·64 | 1·82 | ·38 | 1·55 | 1·74 | ·38 |
| 14 | ... | ... | ... | ... | 1·75 | 1·95 | ·40 | 1·65 | 1·85 | ·40 |
| 15 | ... | ... | ... | ... | 1·87 | 2·08 | ·42 | 1·75 | 1·97 | ·42 |
| 16 | ... | ... | ... | ... | 1·98 | 2·20 | ·44 | 1·85 | 2·08 | ·44 |
| 17 | ... | ... | ... | ... | 2·09 | 2·33 | ·46 | 1·95 | 2·19 | ·46 |
| 18 | ... | ... | ... | ... | 2·20 | 2·45 | ·48 | 2·05 | 2·30 | ·48 |
| 19 | ... | ... | ... | ... | 2·31 | 2·58 | ·50 | 2·15 | 2·41 | ·50 |
| 20 | ... | ... | ... | ... | 2·42 | 2·70 | ·52 | 2·25 | 2·52 | ·52 |

*Note.*—Thicknesses given in above Table are *total* thicknesses, including white-metal, if any.

When double brasses are used the proportions differ slightly from the above, and are approximately,—

**Rule 89e, 89f, and 89g.** $\frac{D}{G}=1{\cdot}25$ ; $\frac{K}{H}=1{\cdot}5$ ; and $\frac{E}{H}={\cdot}47$.

But of course, in this latter case, much depends on the type of crosshead employed, and the side of the jaw should be considered as a cantilever, and the section at S made such that the moment of P does not impose a load of more than 9000 lbs. per square inch, when P is taken as five-eighths of the total effective load on the piston.

## SHAFTING.

The resistance of a plain cylindrical shaft to simple twisting is given by,—

**Rule 90.** $T={\cdot}196\ D^3\times f,$

where T is the twisting moment in inch-lbs. ; D the diameter of the shaft in inches ; and $f$ the greatest shearing stress on the material,—in lbs. per square inch.

When the shaft is hollow, the formula becomes,—

**Rule 90a.** $T={\cdot}196\ \frac{D^4-d^4}{D}\times f,$

where D and $d$ are respectively the external and internal diameters.

The values of $f$ should not exceed the following :—

| | $f$ | |
|---|---|---|
| | Shafts below 10″ dia. | Shafts above 10″ dia. |
| Wrought-iron forging, . | 9,000 | 8,000 |
| Mild steel forging, . . | 12,000 | 11,000 |

For steel of greater strength allow 0·38 × elastic limit in lbs. as the working maximum stress.

**The torsional stiffness of shafts** may be estimated by the following formulæ (Rankine) :—

**Rule 91.** For solid shafts $\theta_r=\frac{10{\cdot}2\ Tl}{CD^4}$ nearly.

**Rule 91a.** For hollow shafts $\theta r=\frac{10{\cdot}2\ Tl}{C(D^4-d^4)}$ nearly.

$\theta_r$ being the measure of angular displacement in *radiants.*

The actual angular movement in degrees taken as $\theta^\circ$, then

For solid shafts $\theta^\circ=\frac{584\ Tl}{CD^4},$

$$\text{For hollow shafts } \theta^\circ = \frac{584\ Tl}{C(D^4 - d^4)},$$

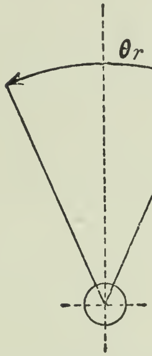


FIG. 14.

$\theta r$ is the length of arc moved through at radius $r$; $l$ length of shaft in inches; C constant, values of which are given below; and other symbols, as defined above.

| | |
|---|---|
| For cast-iron, . . | C = about 2,850,000. |
| ,, wrought-iron, . | C = 8,500,000, to 10,000,000. |
| ,, steel, . . . | C = 10,000,000 to 12,000,000. |

Where a shaft is revolving *uniformly*, and transmitting power, the relation between the twisting moment and the horse-power applied is given by,—

**Rule 92.** $$T = \frac{\text{I.H.P.}}{R} \times 63{,}000.$$

The great number of shafts, however, are turned by steam engines acting through cranks, and do not revolve uniformly,—because the tangential pressure on the crank-pin is constantly varying,—and must therefore be designed to resist the *maximum* twisting moment, instead of the mean given by the above equation.

The rule therefore takes the form,—

**Rule 93.** $$D^3, \text{ or } \frac{D^4 - d^4}{D}, = \frac{\text{I.H.P.}}{R} \times F.$$

Where F is a co-efficient, the values of which depend on the number and relative positions of cranks, distribution of steam in the cylinders, &c., and are given in Tables L. and La.

In Table L. the cranks marked † are at angles of 120°; those marked ‡ at angles of 90°.

## Table L.—Shafts for Screw Engines.

| Description of Screw Engines. | Values of F for | | |
|---|---|---|---|
| | Thrust, etc., Crank-shafts. | Inter-mediate shaft. | Tail-end shaft. |
| Single crank cut off in cylinders not less than 0·5, | 150 | 130 | 175 |
| Two cranks at 90°, cut off in cylinders not less than 0·5 . . . . . . . . | 96 | 84 | 115 |
| Three cranks at 120°, cut off in cylinders not less than 0·5 . . . . . . . | 83 | 72 | 96 |
| Four cranks, balanced triples and quadruples . | 84 | 74 | 96 |
| ,, 90° quadruples . . . . . | 86 | 76 | 98 |
| ,, naval and other high-speed balanced | 74 | 63 | 85 |
| Turbines, direct drive . . . . . . | 70 | 60 | 80 |
| ,, geared generally . . . . . | 70 | 60 | 78 |
| ,, naval, in high-speed light craft . | 37 | 32 | 43 |

## Table La.—Shafts for Paddle Engines.

| Description of Paddle Engines. | Values of F for | | |
|---|---|---|---|
| | Inter-mediate shaft. | Paddle shaft inner journal. | Paddle shaft outer journal. |
| Single crank cut off in cylinder not less than 0·5, | ... | 80 | 100 |
| Two cranks connected by link and equivalent to 90° . . . . . . . . . | ... | 58 | 65 |
| Two cranks at 90° with an intermediate shaft . | 58 | 50 | 65 |
| Three cranks at 120°, solid or built up . . | ... | 45 | 60 |

For ships working in rivers, estuaries, lochs, etc., habitually in smooth water, the value of F may be taken as 80 per cent. of the above.

**Hollow Shafting.**—Shafting for naval ships is now made throughout hollow in order to get the maximum strength and stiffness with minimum weight.

The internal diameter is usually half the external, so that the saving in weight is 25 per cent., whilst the reduction of strength is only about 6¼ per cent.

"Tail" shafts are usually "set in," at the after end, after boring, so as to reduce the diameter of the hole within and near the propeller boss.

## CRANKSHAFTS IN GENERAL.

Crankshafts must be strong enough to bear, together with the maximum torque, the bending and shearing stresses, the magnitudes of which depend on the positions of the bearings. The conditions of loading and support are rather difficult to determine exactly,—depending as they do on the rigidity of the surrounding parts,—but it suffices for most purposes to assume that, in a simple case of overhang, they are represented by the formula,—

**Rule 94.** $$W = \frac{fz}{l},$$

where W is maximum load on pin, in lbs.; $f$ the greatest permissible stress on the material in lbs. per square inch (for which *see* page 113); $l$ length in inches from mid-length of crank-pin to mid-length of bearing; and $z$, for a circular section, is $\cdot 0982\ D^3$.

Where the crank has two arms, and is supported by a bearing on each side, the formula may be taken as,—

**Rule 94a.** $$W = \frac{6fz}{l},$$

$l$ being the distance, in inches, between the *centres* of the two bearings.

**Equivalent Twisting Moment.**—When a shaft is subject to simultaneous twisting and bending, the combined stress on any section of it may be measured by calculating what is called the

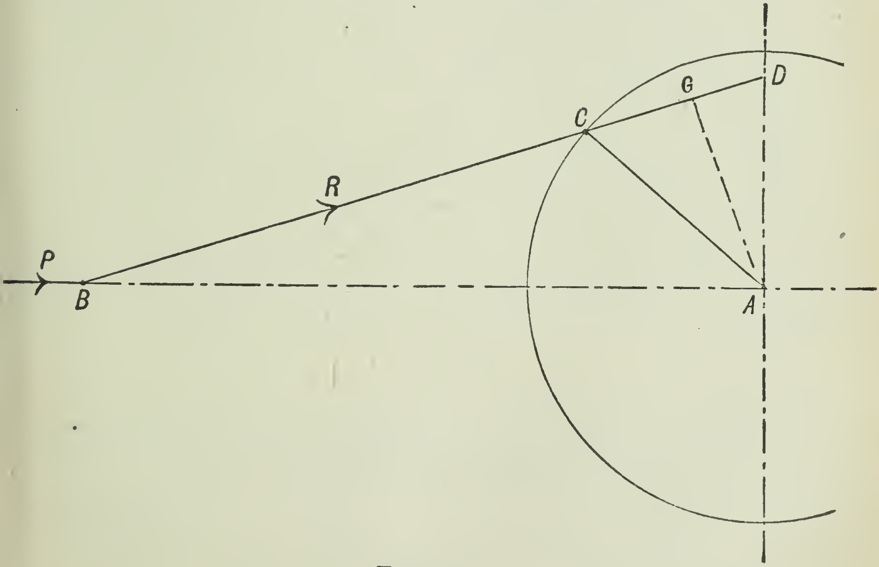


FIG. 15.

equivalent twisting moment; that is, the two stresses may be so combined as to be treated as one twisting stress only, and the size of the shaft calculated accordingly. The formula for combining the stresses is,—

**Rule 95.** $$T_1 = M + \sqrt{M^2 + T^2}$$

where T is the twisting moment; M the bending moment at any section; and $T_1$ the equivalent twisting moment.

The shearing stresses on crankshafts, exclusive of those due to torque, are relatively small, and usually allowed for in the value assumed for $f$ or F.

**Curve of Twisting Moments.**—The twisting moment, at any position of the crank, may be determined graphically as on Fig. 15.

Let AB (Fig. 15) be the centre line of the engine, through the cylinder and shaft centres, AC the position of the crank, BC the connecting-rod, and AD a line at right angles to AB. Produce BC (if necessary) to cut the line AD. Then, if P be the effective pressure on the piston, when the crank is in position AC, the twisting moment is P × AD.

Let the twisting moment be determined at intervals of say 10° of angular movement of the crank, so that there will be 18 values obtained for the half revolutions. Draw a line AB (Fig. 16), and divide it into 18 equal parts, $Aa_1$, $a_1a_2$, &c.; erect perpendiculars at these points, and cut off parts $a_1b_1$, $a_2b_2$, &c., representing to scale the 18 values obtained; and through the points $b_1$, $b_2$, &c., draw a curve, which will be the curve of twisting stress on the shaft during one stroke of the piston. By prolonging AB, and going through a similar operation for the second half of the revolution, the curve of stress during the return stroke may be obtained.

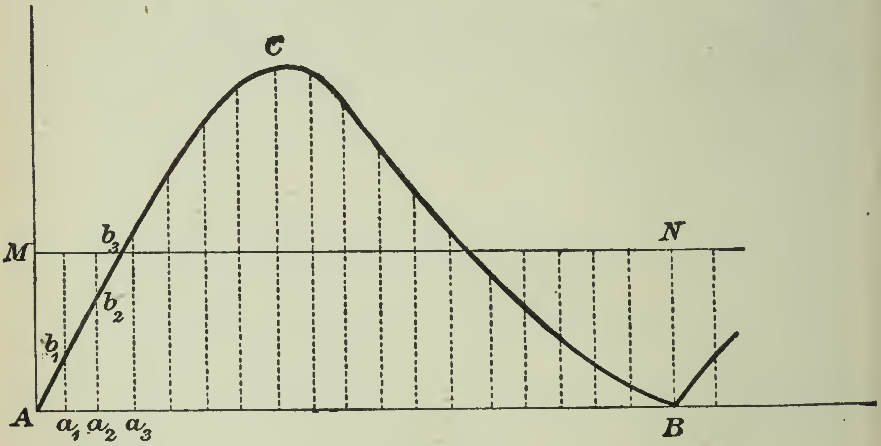


FIG. 16.

If the area enclosed by the curve and the line AB be divided by the length of AB, the quotient (AM in Fig. 16) is the *mean* twisting moment. The value of AM may be calculated by taking a mean of the values of $a_1b_1$, $a_2b_2$, &c. Where there are two or more pistons acting on the same shaft, the curve of combined twisting moments is obtained by laying off the ordinates of the second piston, or crank, above or beyond the first curve, which is used as the base line, and so on, care being taken to step, or displace, the various curves in the direction AB, by an amount due to the angles between the cranks.

The polar form of this diagram, which is preferred by some engineers, is obtained by laying off the ordinates of the various curves radially from a circle which represents the crank-path, instead of from a straight line representing the half crank-path unrolled.

The effective pressures on the pistons at each point are obtained from the indicator diagrams in the manner explained on page 13.

The bending moment on a section of the shaft will vary exactly with the pressure on the crank-pin, and to find the maximum equivalent twisting moment on a section, it is only necessary to construct a secondary curve from the formula $T_1 = M + \sqrt{M^2 + T^2}$, between the point of maximum twisting and that at which the pressure on the piston is greatest.

When steam is not cut off in the cylinder before ·4 of the stroke, the maximum load on the piston may be used to calculate the bending moment which is to be combined with the maximum twisting moment.

**Effect of Inertia of Reciprocating Parts.**—The attention of practical engineers was first drawn to this very interesting subject by Mr Arthur Rigg, in his "Treatise on the Steam Engine," and it has since been treated by various other writers. So far as ordinary marine machinery is concerned, the general effect of this inertia is to equalise the action of the crank in the same manner as a fly-wheel,—the energy expended in accelerating the reciprocating parts during the early part of the stroke being given out again during the latter part, when the steam pressure is lower. When a curve of twisting moments is constructed as described above, and corrected for the effects of inertia, it will generally show that by far the greater part of the steam pressure at the commencement of the stroke is absorbed in producing acceleration, and that the remaining *effective* pressure is but a very small fraction of the whole.

It is not necessary, however, to take these effects into consideration when proportioning the various parts of an ordinary engine, since calculations based on the statical stresses will quite cover all that is necessary to provide for dynamical stresses, except in very extreme cases. For engines of high revolution inertia forces and stresses are not negligible.

**Surfaces of Crank-pins and Main Bearings.**—If the effective bearing surfaces of pins and journals be considered as equal to diameter multiplied by length, then the pressures per square inch should not exceed those given in the following Table:—

### Table LI.—Maximum Working Pressures on Crank-pins, Main Bearings, &c.

| | | |
|---|---|---|
| Crank-pins may carry | $\frac{18,000}{\sqrt{Rd}}$ | lbs. per sq. in. |
| Gudgeons ,, ,, | $\frac{22,600}{\sqrt{Rd}}$ | ,, ,, |
| Main bearings of cranks | $\frac{12,750}{\sqrt{Rd}}$ | ,, ,, |
| Intermediate shaft bearings | $\frac{10,500}{\sqrt{Rd}}$ | ,, ,, |

R being the revolutions per minute.
$d$ the diameters in inches.

The pressures referred to in the above Table are those resulting from the maximum effective loads on the pistons.

## Crankshafts of Screw Engines.

Where a shaft has two, three, or more cranks in it, the after crank has not only to resist the stresses imposed by its own cylinder, but also to transmit the torques produced by all cylinders forward of it, so that,—were it not for practical reasons,—the sizes of crank-arms, journals, and pins, might be progressively reduced from after to forward end of shaft.

On the forward journal and forward arm of forward crank, there is little more torque than is necessary to drive the forward eccentrics (if any), and the bending stresses due to half the load on the pin; whilst the after arm of the same crank, and the journal adjoining it, are subject to similar bending stress, and to nearly the whole of the torque due to the load.

If the bending moment in either journal be expressed by $\frac{M}{2}$, then, the equivalent twisting moment in after journal will be,—according to formula given by Rule 95.

$$T_1 = \frac{M}{2} \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

Coming now to the aftermost crank,—if $T_2$ be the maximum total torque produced by all cylinders forward of it; $T_3$ the maximum torque from its own cylinder; and $M_3$ the corresponding bending moment; then, for the forward journal, the twisting moment is $T_2$, and the bending moment $\frac{M_3}{2}$, so that the equivalent twisting moment is,—

$$T_4 = \frac{M_3}{2} + \sqrt{\left(\frac{M_3}{2}\right)^2 + T_2^2},$$

whilst, on the after journal, the twisting moment is $T_2 + T_3$, and the bending moment $\frac{M_3}{2}$, so that the equivalent twisting moment is,—

$$T_5 = \frac{M_3}{2} + \sqrt{\left(\frac{M_3}{2}\right)^2 + (T_2 + T_3)^2}.$$

The maximum bending moment, acting in a plane at right angles to the axis of the shaft, at any section of the aftermost crank-arm, distant $x$ from the crank-pin centre, is,—

$$M_5 = \frac{T_2 + T_3}{L} \times x,$$

and for the forward arm of the same crank,—

$$M_4 = \frac{T_2}{L} \times x,$$

where L is the length of the crank in inches.

The bending moment at the mid-length of the pin of a solid or rigidly built-up shaft is,—

**Rule 96.** $$M_6 = \frac{R \times L}{8},$$

where R is the load on pin, and L the distance between the centres of the main bearings on either side,—the shaft being considered as a continuous girder supported at the centres of the bearings.

In marine crankshafts, where the crank-arms are not less than ·7 of the diameter in thickness, and where the bearings are close to the crank-arms, there is really little or no bending action at the journals.

At and near the ends of the stroke the crank-arms are subjected to a bending moment acting in the plane of the axis of the shaft and has a magnitude,—

$$M_7 = \frac{R \times L_1}{8},$$

where $L_1$ is the distance from the mid-length of pin to the fore-and-aft centre of the arm (Fig. 17). Hence the thickness ($a$) of the forward arm of forward crank must never be less than,—

$$a = \sqrt{\frac{3R \times L_1}{4bf}}.$$

In ordinary practice, however, to secure interchangeability, the crankshafts of merchant steamers are made with all journals of the same diameter, and all crank-arms of the same dimensions; but in light machinery for Naval purposes the forward crank-arms are sometimes reduced in fore-and-aft thickness.

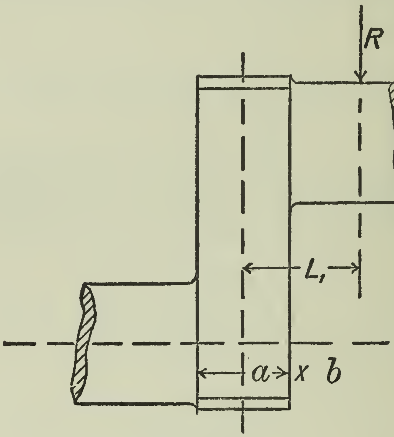


FIG. 17.

Crank-arms, when forged solid with the shaft, will be found to satisfactorily resist all stresses to which they are subject, if made of the following proportions:—

**Rule 97.** **Breadth of arm** = 1·1 × Diameter of shaft.

**Rule 98.** **Thickness of arm** = ·75 × Diameter of shaft.

When for any reason it is not possible to make the arms so thick as ·75 × diameter of shaft, they may be made as thin as ·7, or even, in special cases, ·65; but when this is done, care must be taken to make the breadth such that,—

$$\frac{a \times b^2}{6} = {\cdot}196\ D^3 \times {\cdot}77 = {\cdot}151\ D^3.$$

The most destructive of all stresses to crankshafts are, however, those resulting from the main bearings getting out of line, and the levels of these should therefore be verified every few months, and the shaft lifted and re-bedded whenever any unequal wear is apparent.

The crank-pins of screw engines are nearly always of the same diameter as the shaft journals, and now never less in length than the diameter. The length of the pins, however, will depend on the load per square inch on $(d \times l)$ as allowed by scale in Table LI.

The length of shaft journals cannot, as a rule, be less than the diameter, from practical considerations, and should be such as to comply with scale in Table LI.

To ensure equal wear in the main bearings, they should, theoretically, increase in length from forward aft, in proportion to the twisting moment at each journal,—as the turning force at the crank-pin tends to cause the shaft to move round eccentrically within the bearing and so cause the wear; the after bearing *is* found, in practice, to wear more rapidly than the others, and should not be cut away at the sides, as it has to take the reaction due to the torque of the other cranks as well as that of its own.

**Shaft Couplings and Coupling Bolts.**—(Rule 99.) When couplings are forged solid with the shaft they should have a thickness of ·28 × diameter of shaft.

Coupling bolts should have little beyond a shearing stress to resist, and their size is therefore given by,—

**Rule 100.** **Area of one bolt** $= \dfrac{\cdot196\mathrm{D}^3}{\mathrm{N} \times r}$ or $\dfrac{\cdot196\left(\dfrac{\mathrm{D}^4 - d^4}{\mathrm{D}}\right)}{\mathrm{N} \times r}$ *

if the material of the bolts and that of the shaft offer the same resistance to shearing; D being diameter of shaft, N number of coupling bolts, and $r$ radius at which they are placed.

By way of making some allowance for screwing-up stresses on small bolts, coupling bolts below 1¼-inch diameter may be made 20 per cent. larger than given by above rule.

Where the bolts in any coupling may require to be removed at times, they should be made with a taper of about ¼-inch per foot on the diameters, and be made of harder steel than the shaft.

It is by no means necessary, in large coupling bolts, to make the screwed part equal in diameter to the body of the bolt; bolts between 2½ and 3½ inches diameter may be reduced ¼-inch, and those above 3½ inches, ½-inch in diameter. The nuts may have a thickness of ·75 to ·8 × diameter of screw.

In two, three, and four throw crankshafts, above 10 inch of diameter, each crank, with its adjacent journals, should be made as a separate forging, and the number and position of coupling bolts should be such that the shafts can be interchanged or reversed as may be desired.

---

* *Note.*—Or to obtain diameter at once,—

$$d = \sqrt{\frac{\mathrm{D}^3}{4\,\mathrm{N}\,r}} + \cdot125, \text{ \&c.}$$

When cranks are coupled together in this way, a spigot on each, fitting into a recess in its neighbour, materially reduces the strain on the bolts, assists adjustment, and ensures the "truth" of the shafts when bolted together; such spigots may have a diameter of ·5 to ·7 × diameter of shaft, and a length of say ⅜-inch for the projection, and $^7/_{16}$-inch for the recess.

**Built Crankshafts.**—With a view of obtaining sounder and more reliable work, crankshafts are usually built up of separate pieces. Such shafts are generally cheaper than solid ones, but somewhat heavier.

The thickness of the arms or *webs* are usually somewhat less than those of solid shafts. They should, however, not be less than 0·625 the diameter of the adjacent journal. If $t$ is thickness of the metal around the shank hole, and $h$ the thickness of the web, and $d$ the shank of journal, $t^2 \times h$ should not be less than $0{\cdot}12 \times d^3$. These webs are shrunk on to the shanks and crank-pins; there should be a key or keys on the shaft which are generally cylindrical and sometimes screwed into their holes to prevent end play in case the web gets loose. The diameter of such key should be not less than $0{\cdot}23 \times d + 0{\cdot}2$ inch.

## Crankshafts of Paddle Engines.

These may usually be classed under one of the following three heads:—

(*a*) Intermediate shaft type,—in which the outer crank-arms are keyed upon the paddle-shafts, where the inner ones are similarly fixed to an intermediate shaft, the crank-pins being fixed in the inner crank-arms, and, to a certain extent, free in the outer ones (*v.* fig. 18).

(*b*) The type in which the intermediate shaft and its crank-arms are absent, and the crank-pins are fixed in the paddle-shaft crank-arms, and left long enough to allow of the attachment of a link, or sling, which holds the crank-arms at the required angle with one another.

(*c*) The solid shaft,—in which the cranks are forged in one with the shaft,—as in the case of a screw engine,—and the paddle-shafts connected with it by solid flange couplings, a little elasticity being obtained sometimes by the interposition of a thickness of leather, by helical springs on the coupling bolts, or by other similar devices. Paddle-engine shafts are now generally built up like those of screw engines with the pins of the same diameter as the journals.

The effect of the first arrangement is to communicate very equable stresses to the paddle-shafts, as the pressures from the crank-pins are always at right angles to the cranks on the paddle-shafts; and, in smooth water, the power of each cylinder is very nearly equally divided between the two wheels, and the bending action on each paddle-shaft never exceeds half that due to its own cylinder, for, when near the dead-points, the bending moment is at its maximum, and is wholly taken by the inner crank-arm, to which the pin is secured. For these reasons the intermediate shaft must be stronger than the paddle-shafts, when the ship is intended to work in rough water, as it may have to transmit the whole twisting force from one cylinder, and always takes, during certain parts of the revolution, the whole of the bending forces.

Hence, if T be the maximum twisting moment from one piston of a double cylinder paddle engine, and M the maximum bending moment from the same piston, then,—as in the general case (page 126),—the maximum equivalent twisting moment $T_1$ will be,—

$$T_1 \text{ (on intermediate shaft)} = M + \sqrt{M^2 + T^2}$$

$$\text{and } T_1 \text{ (on paddle-shafts)} = \frac{M}{2} + \sqrt{\left(\frac{M}{2}\right)^2 + T^2}.$$

With the second type of shaft (*b*), the axes of the cylinders are necessarily at an angle with one another, but the cranks are usually so placed that the arrangement is equivalent to one in which the axes are in the same plane and the cranks at right angles.

Each shaft takes the whole of the bending moment from its own cylinder; and,—while usually transmitting half the combined twisting moment—may, in rough water, have to transmit the whole of the twisting moment from one cylinder. The inner journals of these shafts are therefore subject to precisely the same stresses as the intermediate shaft journals in type (*a*), and should be made of the same size, as indicated in Table L.a, page 124.

The maximum pull in the coupling link may be taken as being equal to that in one connecting-rod, but its ordinary value will only be one half of this.

In the third case (that of the solid crankshaft), the cranks will generally be at right angles, and the twisting stresses will be similar to those in the previous cases; the central part transmitting ordinarily half the stress from one piston, but occasionally the whole. The bending stresses in each crank-arm and journal will be those due to half the load on one piston, and may be determined by means of the formulæ previously given.

**Paddle-shafts.**—The outer end of each paddle-shaft is subjected to a bending moment, which is the resultant of those due to the weight of the wheel, and to the reaction of the water on the floats, and the moment of which ($M_1$) may be taken as,—

**Rule 101.** $M_1 = \sqrt{(\text{M of weight})^2 + (\text{M of reaction})^2}.$

The torque at the outer journal of a paddle-shaft is the same as at the inner journal. The outer bearing may have a length of from 1·5 to 2 × diameter of journal, according to service for which the boat is intended, and to weight of wheels.

**Overhung cranks.**—Fig. 18 shows a pair of overhung cranks as formerly fitted in paddle-wheel engines. The crank-pin is subject to bending and shearing stresses due to the thrust on the connecting-rod. The maximum bending stress in the pin is close to the face of the crank and is,—

$$M = R \times l,$$

where R is the thrust on the connecting-rod, and $l$ the length of pin from centre of connecting-rod to face of crank.

The diameter of the pin is given by,—

**Rule 102. Diameter of Crank-pin** $= \sqrt[3]{\frac{R \times l}{f}} \times 10.2,$

where $f$ is the greatest permissible stress on the material, in lbs. per square inch. This rule gives the diameter requisite for strength, but it may be necessary to make the pin larger in order to get sufficient surface to comply with Table LI., page 128.

The crank-arm is to be treated as a lever, so that,—if $a$ is thickness measured parallel to axis of shaft, and $b$ breadth, at a section $x$ inches from the crank-pin,—the bending moment at that section is,—

$$M = R \times x$$

$$\text{and,} \quad \frac{a \times b^2}{6} = \frac{M}{f}$$

$$\text{or } a = \frac{6M}{b^2 \times f}.$$

The bending moment decreases with the distance from the crank-pin, while the shearing force is the same throughout the crank-arm; consequently the section near the crank-pin should have an extra square inch for each 10,000 lbs. of thrust on the connecting-rod, beyond the area necessary to resist the bending moment.

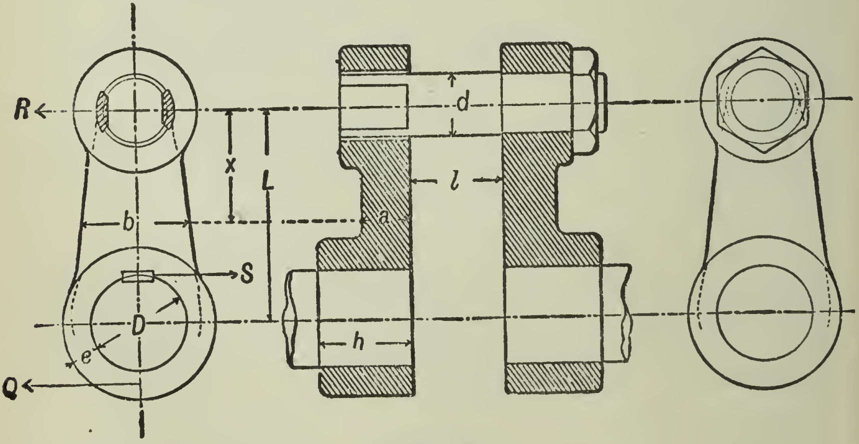


Fig. 18.

The dimensions of the boss to receive the shaft may be determined from,—

**Rule 103.** $$T = 2R\frac{L}{D+e},$$

where T is the total stress on the section $h \times e$, and the other letters have the meanings shown in Fig. 18. The length $h$ is usually ·75 to 1 × diameter of shaft (**Rule 103a**), and,—when crank and shaft are of the same material,—$h$ and $e$ may have the following relative values:—

When $h =$ D then $e =$ ·35D.
,, $h =$ ·9D ,, $e =$ ·38D.
,, $h =$ ·8D ,, $e =$ ·40D.
,, $h =$ ·7D ,, $e =$ ·41D.

The crank-eye, or boss into which the crank-pin is fitted, should bear the same relation to the pin that the shaft boss does to the shaft.

The diameter of the shaft end, on which the crank is fitted, should be 1·1 × diameter of journal.

**Keys for cranks, &c.**—These should be made of steel several grades harder than the shaft and crank, and should be of the following proportions:—

**Rule 104.** **Breadth of Key** $= \frac{D}{4} + {}$·125.

**Rule 104a.** **Thickness of Key** = ·5 × Breadth.

Depth of key-way in shaft, at edge, should be ·17 × Breadth. Some engineers use two (smaller) keys placed 90° apart, in order to obtain at least three bearing points on shaft, and so avoid all risk of "rocking"; and large shafts are sometimes fitted with three keys. Where a lever is placed at some distance from the end of a shaft, so that the key cannot be fitted and driven from the end, it is a good plan to fit a small driving key alongside of the sunk key, as shown in Fig. 19.

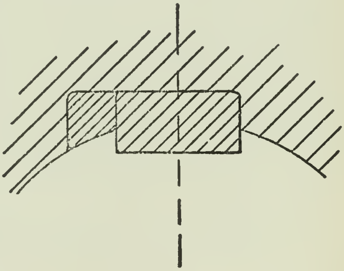


FIG. 19.

## Tail Shafts of Screw Engines.

These must resist the bending action arising from the weight and inertia of the propeller, and from the blades when partially immersed, in addition to the torque. The static stresses can of course be readily determined by the rules already given, but the calculation of the stresses resulting from "racing" of the propeller and pitching of ship is too complex a subject to enter on here. Experience has shown, however, that the tail shaft should have a diameter greater than that of the crankshaft; and it is in practice the rule now to make them considerably larger than the crankshaft, since breakage of one may involve serious consequences. Much also depends on the manner in which the tail-shaft is supported.

**Rule 105.**—The taper of the end that fits into the propeller boss should not be less than ¾-inch per ft., and is generally 1 inch per ft.

The thread of the large nut that holds the propeller on the shaft is commonly made 2½ threads per inch, regardless of the diameter; it should be left-handed when the propeller is right-handed, and *vice versa.* The nut should be securely locked, preferably by a plate fixed to the after end of the boss by tap screws.

The propeller should be secured by one feather or key extending the whole length of the boss, the proportions of which may be,—

**Rule 106.—Breadth of key** = ·22 × largest diameter of shaft + ·25.

**Rule 106a.—Thickness of key** = ·55 × breadth.

The diameter of the screwed end of the shaft should be sufficiently reduced to allow of the key being fitted in, from the after end, clear of the thread.

The diameters over the brass casings at the bearings should differ by ⅛ inch, to enable the shaft to be got in and out of the stern-tube more easily. The thickness of brass casing should not be less than,—

**Rule 107.—Thickness of brass casing at bearings** = $\cdot 055 D_1 + \cdot 15$ where $D_1$ is the diameter of the shaft at the bearing, within the casing.

See Table LV., page 144.

## Board of Trade Regulations for Shafting (1918).

Main, tunnel, propeller, and paddle shafts should not be passed if less in diameter than that found by the following formulæ, without previously submitting the whole case to the Board of Trade for their consideration. First-class makers generally put in somewhat larger shafts than those obtained by the formulæ.

For compound condensing engines with two or more cylinders, when the cranks are not overhung :—

**Rule 108.**

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}.$$

**Rule 108a.**

$$P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right).$$

Where $S$ = diameter of shaft in inches.

$d^2$ = square of diameter of H.P. cylinder, in inches; or sum of squares of diameters when there are two or more H.P. cylinders.

$D^2$ = square of diameter of L.P. cylinder, in inches; or sum of squares of diameters when there are two or more L.P. cylinders.

$P$ = *absolute* pressure in lbs. per square inch,—that is, boiler pressure plus 15 lbs.

$C$ = length of crank in inches.

$f$ = constant from following table.

## Rules for Shafting of the British Marine Engineering Design and Construction Committee.

(i.) *For Turbine Shafts.*—Diameter = $\sqrt[3]{\frac{\text{S.H.P.} \times F}{R}}$.

For the intermediate shafts of ocean-going ships F = 64.
,, ,, smooth-water service generally . . F = 58.

The tail-end shaft at outer end near propeller diameter = diameter of intermediate shaft + $\frac{\text{diameter of screws}}{120}$.

(ii.) *For Reciprocating Steam Engines* whose L.P. cylinder is D and the stroke of piston S, both in inches. The ratio of the L.P. to the H.P. cylinder is $r$ and the working pressure P.

$$\text{Diameter of intermediate shafts} = \sqrt[3]{\frac{D^2 \times S \times P}{K(r+2)}}.$$

For engines with two cranks at 90°, cut off not earlier than 0·5, F = 1800.
,, ,, ,, three ,, ,, 120°, ,, ,, ,, F = 2000.
,, ,, ,, four ,, balanced specially, F = 2000.

For ships on home service, short voyages, and for smooth-water service F may be taken as 10 per cent. larger.

The thrust-shaft near collars, the crankshaft, and the inner end of tube shaft to be in all cases 5 per cent. larger in diameter than the intermediate.

## For the Shafting of Turbines.

$$\text{Board of Trade Rule diameter} = \sqrt[3]{\frac{\text{I.H.P.} \times f}{R}}.$$

I.H.P. is the estimated *maximum* power transmitted through the shaft at R revolutions per minute.

$f = 60{\cdot}3$ for tunnel shafts. $f = 82{\cdot}5$ for propeller shafts.

## Table LII.—Board of Trade Constants for Shafts.

| For two Cranks. Angle between Cranks.* | For Crank and Thrust Shafts.† | | For Tunnel Shafts. | For Propeller Shafts. |
|---|---|---|---|---|
| 90° | | 1,047 | 1,221 | 890 |
| 100° | For paddle engines multiply constant in this column suitable by 1·4. | 966 | 1,228 | 821 |
| 120° | | 855 | 997 | 727 |
| 140° | | 788 | 919 | 670 |
| 160° | | 751 | 877 | 638 |
| 180° | | 740 | 864 | 629 |
| Three Cranks, 120° | | 1,110 | 1,295 | 943 |

* When there is only one crank, the constants applicable are those in the Table, opposite 180°.

† The diameter of the tail shaft can be found by multiplying the diameter of the intermediate by 1·112 in every case.

## Lloyd's Rules for Shafting.

The diameters of *intermediate* shafts are to be not less than those given by the following formulæ :—

### Table LIII.—Diameters of Shafts.

| Description of Engine. | Diameter of Intermediate Shaft in inches. |
|---|---|
| Compound—Two cranks at right angles . . | $(\cdot04A+\cdot006D+\cdot02S)\times\sqrt[3]{P}$. |
| Triple — Three cranks at equal angles . . | $(\cdot038A+\cdot009B+\cdot002D+\cdot0165S)\times\sqrt[3]{P}$. |
| Quadruple—Two cranks at right angles . . | $(\cdot034A+\cdot011B+\cdot004C+\cdot0014D+\cdot016S)\times\sqrt[3]{P}$. |
| Do. —Three cranks . . | $(\cdot028A+\cdot014B+\cdot006C+\cdot0017D+\cdot015S)\times\sqrt[3]{P}$. |
| Do. —Four cranks . . | $(\cdot033A+\cdot01B+\cdot004C+\cdot0013D+\cdot0155S)\times\sqrt[3]{P}$. |

Where A is diameter of H.P. cylinder in inches.
B ,, first I.P. cylinder in inches.
C ,, second I.P. ,,
D ,, L.P. ,,
S stroke of pistons in inches.
P boiler pressure above atmosphere in lbs. per square inch.

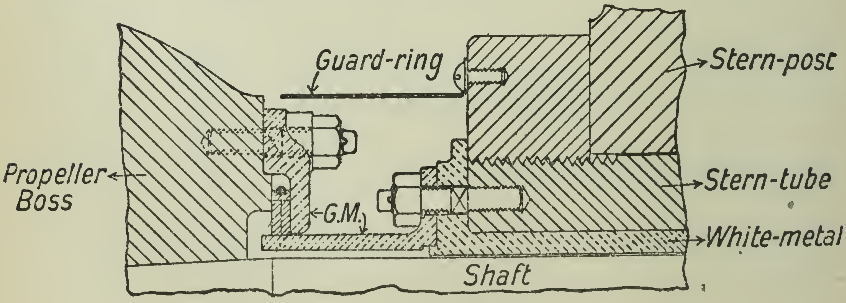


FIG. 20.

Diameter of crankshaft, and of thrust-shaft under collars, to be at least $\frac{21}{20}$ths of that of intermediate shaft; thrust-shaft may be tapered down at each end to same diameter as intermediate shaft.

Diameter of screw (tail) shaft to be equal to diameter of intermediate shaft $\times\left(\cdot 63+\frac{\cdot 03\ P}{T}\right)$, but in no case to be less than $1\cdot 07\ T$, where P is diameter of propeller and T diameter of intermediate shaft, both in inches.

Fig. 20 shows an arrangement of oil-lubricated stern bearing suggested by the authors as simple and efficacious. The oil is supplied from a tank above the water-line.

## THRUST BLOCKS.

To find the thrust along the shaft of a screw engine, it is necessary to know the speed of the ship, and the effective horse-power delivered by the screw, that is, S.H.P. minus the power lost by friction, etc., of the screw itself.

The effective horse-power is the power actually employed in propelling the ship, and its relation to the indicated horse-power depends therefore on the combined efficiency of the engines and propeller.

For the purpose of calculating the surface of thrust-collars it is sufficient to assume the E.H.P. ÷ I.H.P. = K when

| | |
|---|---|
| For turbine engines of best make, . . . . . . . . | $K=0\cdot 80$ |
| ,, best fast-running vertical reciprocators with no pumps, . | $K=0\cdot 77$ |
| ,, such engines with air pumps, . . . . . . . . | $K=0\cdot 75$ |
| ,, good engines of medium speed having air, feed, and bilge pumps, | $K=0\cdot 71$ |
| ,, ordinary mercantile engines with such pumps, . . . . | $K=0\cdot 68$ |

Then,—

**Rule 109. Indicated Thrust in lbs.** $=\frac{\text{I.H.P.}\times 33{,}000}{S\times 101\cdot 3}=\frac{\text{I.H.P.}\times 326}{S}$,

where S is the speed of the ship in knots; and,—

**Rule 110. Mean Normal Thrust in lbs.** $=\frac{\text{I.H.P.}\times 326\times K}{S}$.

As the thrust varies with the I.H.P., and inversely with the speed (*see* Table LIV.), it may rise considerably above the "mean normal" value, if, from any cause, the speed is reduced without a corresponding decrease in the power, as, for instance, when "thrashing" against a head wind and sea, or when towing.

**Rule 111.** The pressure per square inch, due to the mean normal thrust, should be about $(2700\div\sqrt{Rd}+100)$ lbs., R being revolutions per min. and $d$ diameter of thrust-shaft in inches; for tug boats, it should not exceed 50 lbs.

In practice the pressure is about 50 lbs. in naval work, and about 60 lbs. in mercantile steamers, when white metal is used; with the Michel thrust (fig. 21) it may be as high as 500 lbs. with a coefficient of friction 0·0015 only. In practice 300 to 400 lbs. is usual.

The relation between depth and thickness of collars on shaft is given by,—

**Rule 112.** $$\text{Thickness} = \begin{cases} \cdot 25\,(D - d) + \cdot 4 \text{ inch (with small collars),} \\ \cdot 17\,(D - d) + \cdot 4 \text{ inch (with large collars),} \end{cases}$$

where D is diameter over collars, and $d$ diameter between collars,—both in inches.

The spaces between collars on shafts should be as follows:—

**Rule 113.** When rings in block are solid gun-metal. } $\text{Space} = \cdot 4\,(D - d).$

**Rule 113a.** When block is fitted with cast-iron or cast-steel "horse-shoes" faced with brass or white metal, as shown in fig. 22. } Cast-iron,— $\text{Space} = 1{\cdot}8\sqrt{(D - d) - 1{\cdot}5}.$ Cast-Steel,— $\text{Space} = 1{\cdot}5\sqrt{(D - d) - 1{\cdot}5}.$

**Rule 113b.** When rings are cut out of solid metal of block and cap, and entire corrugated surface is covered with uniform layer of white metal. } $\text{Space} = \cdot 4\,(D - d) + \cdot 6 \text{ in.}$

The number of collars employed is, to some extent, optional. With the Michel thrust block one is sufficient and usual, but for very large power a second can be added to get the necessary surface with a reasonable diameter. With the ordinary horse-shoe thrust it is usual to have one collar for shafts up to 6 inches diameter, and over that an additional collar for each 2 inches of diameter, or such a number as will give the necessary surface with a moderate diameter.

The shaft should be supported and "steadied" in its place so as to bear properly on the horse-shoe collars without any tendency to oscillate and cause unfairness in contact. To attain this there should be a bearing on each side of the block when it is a large and long one, and for small shafts with a short block one will serve the purpose, and it may be in one with the block.

The lubrication must, of course, be certain and plentiful and applied at the right places for efficiency; the collars should really run in an oil-bath so as to be self-lubricative. In the case of the Michel thrust the lubrication is perfect, and consequently with this system the pressure necessary to squeeze out the oil-film interposed between the rubbing surfaces is very high. Large collars of type fig. 22 may be made hollow with advantage and then fitted with water circulation to carry away the heat generated. To facilitate examination and refit the thrust shaft should be a short one.

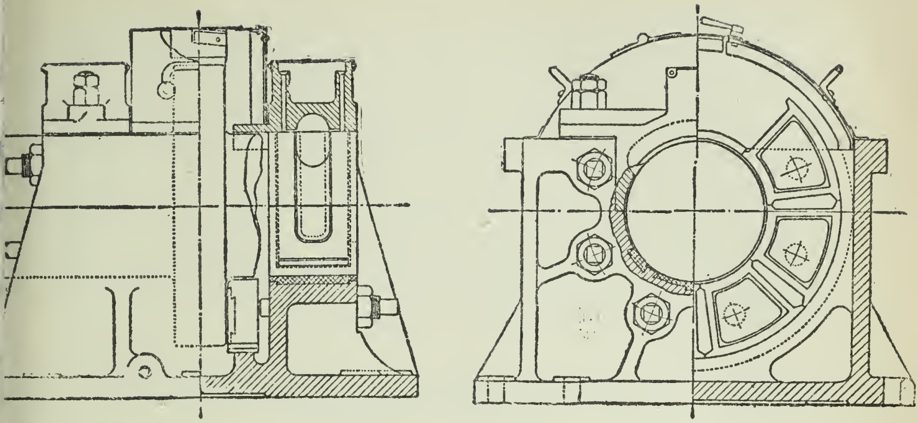
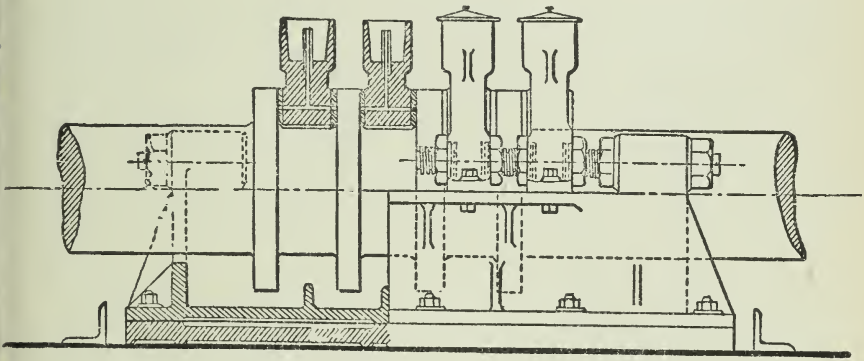


FIG. 21.—Michel Block.



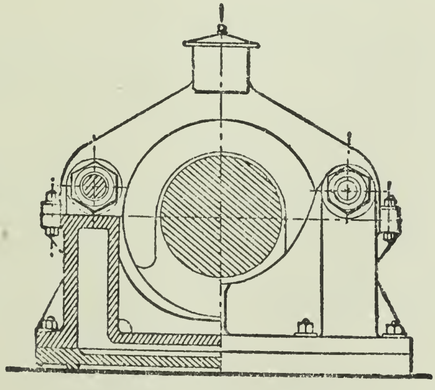


FIG. 22.—Thrust Block.

Thrust-blocks of all types should be carefully scraped and bedded to a perfect bearing on the shaft collars before leaving the workshops.

**Rule 114.** The number and diameter of holding-down bolts should be such that the shearing stress per square inch, at the bottom of thread due to the mean normal thrust, may be between 1800 and 2200 lbs., when the bolts are unassisted by stops or angle bars riveted to the top plate of the seating, and between 3200 and 3800 lbs. when such stops are fitted. Stops should always be fitted where possible, as, when they are absent, the stress on the bolts is largely a bending stress, owing to the unavoidable presence of a certain thickness of packing.

The sizes of bolts usually employed vary from ⅞ inch to 1½ inch.

Packing should be of cast-iron, *not* of wood.

### Table LIV.—Thrust Surfaces per I.H.P. for Various Speeds of Vessels.*

| Speed in knots. | Surface necessary to keep pressure 50 lbs. per sq. inch. | Speed in knots. | Surface necessary to keep pressure 50 lbs. per sq. inch. | Speed in knots. | Surface necessary to keep pressure 50 lbs. per sq. inch. | Speed in knots. | Surface necessary to keep pressure 50 lbs. per sq. inch. |
|---|---|---|---|---|---|---|---|
| | Sq. inch. | | Sq. inch. | | Sq. inch. | | Sq. inch. |
| 8 | ·540 | 15 | ·288 | 22 | ·198 | 29 | ·150 |
| 9 | ·485 | 16 | ·270 | 23 | ·190 | 30 | ·145 |
| 10 | ·436 | 17 | ·255 | 24 | ·182 | 31 | ·141 |
| 11 | ·396 | 18 | ·240 | 25 | ·175 | 32 | ·137 |
| 12 | ·361 | 19 | ·227 | 26 | ·168 | 33 | ·133 |
| 13 | ·335 | 20 | ·216 | 27 | ·161 | 34 | ·129 |
| 14 | ·312 | 21 | ·207 | 28 | ·155 | 35 | ·125 |

## STERN-TUBES.

In merchant vessels the stern-tube is almost invariably of cast-iron, fitted with a brass bush, or bushes, to carry the lignum-vitæ strips which form the actual bearing surface for the shaft.

In Naval vessels, on the contrary, the stern-tube is almost always of gun-metal, and the bushes are not always fitted, as the lignum-vitæ strips may be fitted into grooves in the tube itself.

In multiple-screw Naval vessels the shaft brackets are fitted with gun-metal bushes, carrying strips of lignum-vitæ,—very similar to those used for the after ends of stern-tubes in merchant vessels; and the forward ends of the stern-tubes are also fitted with long lignum-vitæ bearings, for the purpose of reducing vibration in the unsupported outboard shafts.

The stern-tubes of merchant vessels should be secured in place by a ring nut on the after end, made to screw up against the after face of the stern-post and draw the tube aft until a collar, formed on it, is close up to forward side of post; the forward end of tube being, of course, jointed to the collision bulkhead in the usual manner.

---

* With Michel thrust block allow one-seventh of these surfaces.

Naval stern-tubes do not require any fixing at the after end, as they are always surrounded by steel tubes, ½- to ⅝-inch thick, which tie the collision bulkhead to the post, or, in the case of twin-screw ships, to the special stern-tube brackets; they are simply drawn tightly into the post, or brackets, and bolted to the bulkhead at the forward end. When, as occasionally happens, they are put in from the after end, the flange at the stuffing-box end is made separately and screwed on.

Cast-iron stern-tubes should be of the thickness given by,—

**Rule 115.** $$T = \frac{D}{12} + \cdot 5 \text{ in.},$$

and gun-metal tubes and bushes for shaft-brackets by,—

**Rule 115a.** $$T = \frac{D}{22} + \cdot 35 \text{ in.},$$

where D is the greatest diameter of shaft, over casings. (*See* Table LV., page 144).

The stern-bush, for either shaft-bracket or stern-tube, should be of such length that the pressure per square inch (taking surface as length × diameter) does not exceed 35 lbs., and a good rule is,—

Length of bush is usually 3 to 4 × diameter of shaft,* over casings.

This rule will be found to limit the pressure to under 30 lbs. per square inch, but it should be in accordance with,—

**Rule 116.** $$\text{Pressure per sq. inch} = \frac{1250}{\sqrt{Rd}}.$$

The bush at forward end of stern-tube should not be shorter than one diameter for single screw vessels, and, for twin-screw vessels, where there is a length of unsupported shaft outside the stern-tube, the length of this bush should not be less than 1·5 to 1·75 × diameter of shaft.

These bushes in stern-tubes should have a thickness of metal at the back of the lignum-vitæ not less than,—

**Rule 116a.** $$T_1 = \frac{D}{28} + \cdot 25 \text{ in.},$$

where D is diameter of shaft over casing.

The ribs between the lignum-vitæ strips may have a thickness of ⅜-inch for a shaft 6 to 8 inches in diameter, increasing to ¾-inch for a shaft of 18 inches diameter.

The lignum-vitæ strips themselves may vary from ½-inch × 2 inches to 1 or 1⅛ inch × 4 inches, for shafts of the diameters just mentioned.

The stuffing-box and gland at the inner end of stern-tube may have the following proportions :—

---

* Lloyd's now require 4 diameters.

Rule 117.

$$\text{Depth of stuffing-box} = \frac{D}{2} + 1{\cdot}5 \text{ ins.}$$

Rule 117a.

$$\text{Amount gland can enter} = \frac{D}{5} + 1 \text{ in.}$$

Rule 117b.
Diameter of stuffing-box $= (1{\cdot}1 \times D) + 1{\cdot}5$ ins. for bushed glands.
Rule 117c.
Diameter of stuffing-box $= (1{\cdot}06 \times D) + 1{\cdot}5$ ins. for solid G.M. glands.

The gland studs may vary from four, 1 inch in diameter, for a shaft of 6 to 8 inches diameter, up to eight, $1\frac{1}{4}$ inch diameter, for a 16- or 18-inch shaft. If pinion nuts and toothed rings are used, the gland studs must, of course, be made stouter to stand the bending stresses, and should then vary from $1\frac{1}{4}$ inch to $1\frac{5}{8}$ inch.

The following Table shows at a glance the thicknesses of shaft casings, stern-tubes, and stern-bushes, and the dimensions of stern-tube stuffing-boxes, as calculated by Rules 107, 115, 116a, 117 :—

## Table LV.—Stern-Tubes and Bushes, etc.

| Dia. of shaft under casing. ($D_1$) | Dia. of shaft over casing. (D) (casing $= {\cdot}055D_1 + {\cdot}15$) | Thickness of cast-iron tube. $\frac{D}{12} + {\cdot}5$ | Thickness of gun-metal tube or shaft-bracket bush. $\frac{D}{22} + {\cdot}35$ | Thickness of tube bushes at back of lignum-vitæ. $\frac{D}{28} + {\cdot}25$ | Depth of stuffing box. $\frac{D}{2} + 1{\cdot}5$ | Amount gland can enter. $\frac{D}{5} + 1$ | Dia. of stuffing box for bushed cast-iron gland. $1{\cdot}1\,D + 1{\cdot}5$ | Dia. of Stuffing box for solid gun-metal gland. $1{\cdot}06\,D + 1{\cdot}5$ |
|---|---|---|---|---|---|---|---|---|
| 6 | 7 | $1\frac{1}{16}$ | $\frac{11}{16}$ | $\frac{1}{2}$ | 5 | $2\frac{3}{8}$ | $9\frac{1}{4}$ | 9 |
| 7 | 8 | $1\frac{3}{16}$ | $\frac{3}{4}$ | $\frac{1}{2}$ | $5\frac{1}{2}$ | $2\frac{5}{8}$ | $10\frac{1}{4}$ | 10 |
| 8 | $9\frac{1}{8}$ | $1\frac{1}{4}$ | $\frac{3}{4}$ | $\frac{9}{16}$ | 6 | $2\frac{7}{8}$ | $11\frac{1}{2}$ | $11\frac{1}{8}$ |
| 9 | $10\frac{1}{4}$ | $1\frac{3}{8}$ | $\frac{13}{16}$ | $\frac{5}{8}$ | $6\frac{5}{8}$ | 3 | $12\frac{3}{4}$ | $12\frac{3}{8}$ |
| 10 | $11\frac{3}{8}$ | $1\frac{7}{16}$ | $\frac{7}{8}$ | $\frac{11}{16}$ | $7\frac{1}{8}$ | $3\frac{1}{4}$ | 14 | $13\frac{1}{2}$ |
| 11 | $12\frac{1}{2}$ | $1\ \frac{\ }{16}$ | $\frac{15}{16}$ | $\frac{11}{16}$ | $7\frac{3}{4}$ | $3\frac{1}{2}$ | $15\frac{1}{4}$ | $14\frac{3}{4}$ |
| 12 | $13\frac{5}{8}$ | $1\frac{5}{8}$ | $\frac{15}{16}$ | $\frac{3}{4}$ | $8\frac{1}{4}$ | $3\frac{3}{4}$ | $16\frac{1}{2}$ | 16 |
| 13 | $14\frac{3}{4}$ | $1\frac{3}{4}$ | 1 | $\frac{3}{4}$ | $8\frac{7}{8}$ | 4 | $17\frac{3}{4}$ | $17\frac{1}{8}$ |
| 14 | $15\frac{7}{8}$ | $1\frac{13}{16}$ | $1\frac{1}{16}$ | $\frac{1\ }{16}$ | $9\frac{3}{8}$ | $4\frac{1}{4}$ | 19 | $18\frac{3}{8}$ |
| 15 | 17 | $1\frac{15}{16}$ | $1\frac{1}{8}$ | $\frac{7}{8}$ | 10 | $4\frac{3}{8}$ | $20\frac{1}{4}$ | $19\frac{1}{2}$ |
| 16 | 18 | 2 | $1\frac{3}{16}$ | $\frac{7}{8}$ | $10\frac{1}{2}$ | $4\frac{5}{8}$ | $21\frac{3}{8}$ | $20\frac{1}{2}$ |
| 17 | $19\frac{1}{8}$ | $2\frac{1}{8}$ | $1\frac{3}{16}$ | $\frac{15}{16}$ | 11 | $4\frac{7}{8}$ | $22\frac{1}{2}$ | $21\frac{3}{4}$ |
| 18 | $20\frac{1}{4}$ | $2\frac{3}{16}$ | $1\frac{1}{4}$ | 1 | $11\frac{5}{8}$ | 5 | $23\frac{3}{4}$ | 23 |
| 19 | $21\frac{3}{8}$ | $2\frac{1}{4}$ | $1\frac{5}{16}$ | 1 | $12\frac{1}{8}$ | $5\frac{1}{4}$ | 25 | $24\frac{1}{8}$ |
| 20 | $22\frac{1}{2}$ | $2\frac{3}{8}$ | $1\frac{3}{8}$ | $1\frac{1}{16}$ | $12\frac{3}{4}$ | $5\frac{1}{2}$ | $26\frac{1}{4}$ | $25\frac{3}{8}$ |

In designing gun-metal stern-tubes, shaft casings, &c., great care must be taken to avoid, as far as possible, all shoulders or steps in the longitudinal section, as the castings set so rapidly, and consequently grip the core so quickly, that everything tending to hinder the longitudinal contraction is a source of danger; want of attention to this will probably cause a long tube to be torn into two or three separate pieces in cooling. The middle portions of Naval stern-tubes are commonly made about 30 per cent. thinner than the ends or portions containing the bearings. In torpedo boats, &c., the tube-shafts seldom have any casings, and run in white metal bearings.

## MAIN BEARINGS.

The surfaces and lengths of main bearings have already been dealt with.

**Main Bearing Bolts.**—These should be of "mild" steel, with iron nuts, and, to allow for variations in adjustment, &c., should be proportioned as follows:—

**Rule 118.** When there are two main bearings to each crank, and two bolts to each cap,—assume that each bolt must carry one-third of maximum effective load on piston, and select the proper diameter from Table XLIII., page 103.

**Rule 118a.**—When there are two bearings on each crank, but four bolts to each cap,—assume that each bolt must carry one-fifth of maximum effective load on piston, and select diameter from same Table.

**Rule 118b.**—Where there is only one long bearing with four cap bolts, between two cranks of a triple engine (cranks at 120°),—assume that each bolt must carry one-fourth of maximum effective load on piston and select diameter as before.

**Rule 118c.**—When the crank is overhung, as in various types of paddle-engine, and the whole thrust of the piston-rod, at the ends of the stroke, is resisted by two cap bolts,—call the thrust P; the distance from centre of piston-rod to centre of main bearing bolts *pw*; and the distance from centre of piston-rod to centre of outer bearing (in case of paddle-shaft), or of bearing at other end (in case of intermediate shaft) *pf*. Then (fig. 23),—

$$\textbf{Stress on main bearing bolts} = P \times \frac{pf}{wf}$$

and each bolt must carry one-half of this.

**Rule 118d.** When the conditions are similar to (**c**), but there are four bolts in each cap,—assume that each bolt must carry one-third of total load on cap, and select suitable diameter from Table XLIII., page 103.

In cases (**c**) and (**d**) the stresses on the main bearing bolts due to the weight and reaction of the wheels should be calculated on the principle indicated above, and allowed for if necessary; their magnitude of course depends very greatly on the length of the paddle-shaft.

**Main Bearing Caps or Keeps.**—In order to allow for variations in adjustment, either unnoticed or such as occur when a bearing warms up and has to be slacked, the loads on main bearing caps may be assumed to be as follows :—

**Rule 119.**

**Two bearings to each crank** (cases 118 and 118a) . . . . . . . . . . . . . $\frac{2}{3}$ × **max. effect. load on piston.**

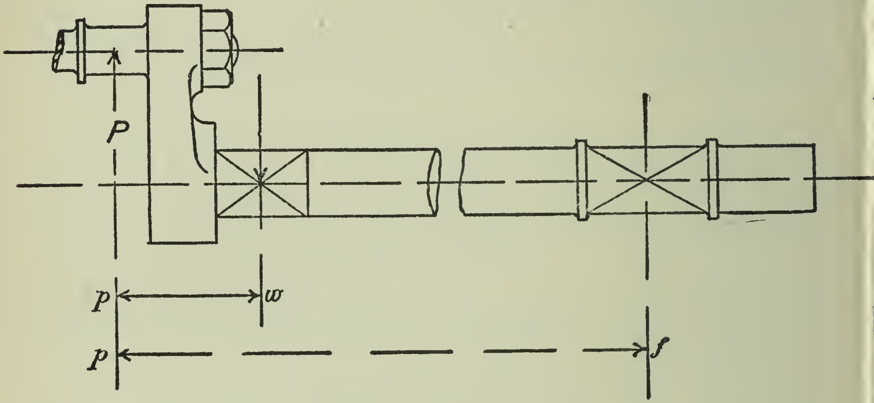


FIG. 23.

**Rule 119a.**

**One long bearing between two cranks** (case 118b) . . . . . . . . . . . . $\frac{3}{4}$ × **max. effect. load on piston.**

**Rule 119b.**

**One bearing to each crank** (cases 118c and 118d) . . . . . . . . . . . . $\frac{pf}{wf}$ × **max. effect. load on piston.**

The load on a main bearing cap is neither a single central load, nor is it a uniformly distributed load, though probably nearer the latter than the former. If suitable values for the working stresses are employed, the most generally convenient formula is that for a beam supported at ends, and uniformly loaded, viz. :—

**Rule 120.** $$w=\frac{8fz}{l^2},$$

where $w$ = load in lbs. per inch of span ($l$);
$l$ = span, or distance from centre of cap bolt to centre of cap bolt;
$z$ = Modulus of section $\left(=\frac{\text{breadth} \times \text{depth}^2}{6}\right.$ for a rectangular section; for other sections see Table CXXXVII.);

and $f$ has the following values :—

| | | |
|---|---|---|
| **Flat-backed brasses and caps** | Cast-iron . . . . | 2,500 |
| | Wrought-iron or "mild" steel | 9,000 |
| | Cast-steel . . . . | 10,000 |
| **Half-round brasses and caps** | Cast-iron . . . . | 3,000 |
| | Wrought-iron or "mild" steel | 10,000 |
| | Cast-steel . . . . | 11,000 |

When the caps are made of cast-steel the section at the centre may, with advantage, be made to approximate to that of a "channel" bar by employing two deep external ribs; but in this case the value of $f$ must not exceed 6000. The values given above for cast-steel are no doubt rather low, but there is,—in the present state of the manufacture,—always some risk of unsoundness, blow-holes, &c.

In estimating the breadths of caps care should be taken to deduct the breadths, or diameters, of all hand holes, oil holes, &c.

**Main Bearing Brasses.**—For small engines, auxiliary engines, &c., the inner or under brass may be of a semi-octagonal form, and the outer or upper one flat on the back; but in larger engines the under brass at least should be of the "half-round" type, and, when the framing can be arranged to allow of it, it is both cheaper and better to make the upper one "half-round" also. Some engineers, however, prefer the square recess with the flat bottom brass which can be packed up to keep the shaft bearings in true line when required.

The over-all thickness of the brasses (*i.e.* the thickness including white metal) should be as follows:—

**Rule 121.** **Flat-backed brasses** **Thickness** $= \frac{D}{9} + {\cdot}3$-inch.

**Rule 121a.** **Round brasses . .** **Thickness** $= \frac{D}{10} + {\cdot}25$-inch.

**Rule 121b.** **White metal . . .** **Thickness** $= {\cdot}02\,D + {\cdot}125$ inch, where D is diameter of journal in inches.

The recesses in the brasses should be carefully tinned before the white metal is run in, and the practice of hammering the white metal to consolidate it should be avoided.

In large engines of the mercantile marine the shells for white metal are commonly and are better made of strong cast-iron, and sometimes the caps and shell upper half are cast in one.

When no white metal is used the thickness of brass at crown should be,—

**Rule 122.**

**Thickness of brass (when no white metal)** $= \begin{cases} \frac{D}{10} + {\cdot}15 & \text{when flat-backed.} \\ \frac{D}{11} + {\cdot}12 & \text{when round.} \end{cases}$

Brasses should never overhang the frames or caps by more than "thickness of brass," as given by above rules, at each end.

See Table XLIX. for the proportions given by the above rules.

The tendency of brasses to close, after being warm, and grip the shaft, should be provided against either by securing them to the frames by screws or bolts, or, a simpler and better way, by H section liner strips.

In large main bearings it is very useful to have the lock-rings of the nuts graduated, in order that, after being slacked back, the nuts may be returned exactly to the old positions.

## FRAMINGS.

The section of the girder that carries the main bearing is usually either of the "H" or "box" type. For small engines either the H or open bottom box is the best type, and for large engines the box,—if the frame is of cast-iron,—or the H,—if cast-steel is used.

In the case of vertical screw engines for Naval purposes this girder is sometimes cut down to the very slenderest proportions, so far as the engine builder is concerned, and is built of steel plates and angles as a part of the engine seating. In oscillating paddle-engines, on the other hand, the girder receives no support from any seating, and must be strong enough to deal with all stresses communicated to it from the bearing. Between these two extremes almost every possible intermediate case is found in practice, and the designer must meet them by using one or other of the general formulæ for girders, and must use his judgment in deciding what co-efficients best express the conditions of loading, supporting or fixing, &c.,—not forgetting that the material is subject to alternate tensile and compressive stresses.

In the worst case (that of the oscillating paddle-engine) the girder should be taken as *supported* only, and the load considered as a single central one, and the values of $f$ may be 3000 for cast-iron and 8000 for cast-steel. In this particular example there are frequently other stresses,—due to the unsymmetrical positions of the columns or pillars with regard to the bearings,—which must also be allowed for.

In vertical screw engines there are stresses on these girders communicated from the columns, and due to the thrusts on the guides, and also to the weight of the cylinders, &c., when the ship is rolling.

In paddle-wheel vessels there are also (as previously mentioned) stresses due to the weights and reactions of the wheels, &c.

Very valuable additional strength and stiffness may sometimes be obtained by using a forged or cast-steel cap with lips at the ends which prevent the springing open of the gap containing the brasses, or, spigots or projections may be used which will prevent it either opening or closing.

## COLUMNS.

The tensile stresses on the material of columns, and also on the bolts connecting them to the cylinders have been dealt with under the heading "Column feet and Bolts," page 97 ; but in vertical engines there are also bending stresses due to the thrusts on the guides, and to the weight of the cylinders, pistons, valves, rods, &c., when the ship is rolling. These may all be dealt with by means of the formula for a beam fixed at one end and loaded at the other, viz. :—

$$W = \frac{fz}{l},$$

and need not therefore be considered in detail here.

The bolts attaching the column feet to the bed-plate or frame must of course be strong enough to carry the direct tensile stress, plus the stress caused by the tendency of the column to cant over on one edge of its base; and it must be noticed that only about one-fourth of the bolts in each column foot are, on an average, able to offer resistance to the latter stress.

When weight is of great importance, as in Naval work, and framings are made of the lightest possible sections, special stresses,—such as those resulting from the action of the reversing engine, &c.,—must be carefully considered and allowed for.

## CONDENSERS.

### 1. Jet Condensers.

**The Capacity** of a jet condenser should not be less than one-fourth that of the cylinder, or cylinders, exhausting into it, and need not be more than one-half,—unless the engine is a very quick-running one: one-third the capacity of cylinder is generally sufficient.

**The Form** of a jet condenser is not a matter of much consequence, and depends, to some extent, on the type and design of engine to which it is fitted. The inlet for steam should be high enough to prevent water getting into the cylinders, and the lower portion should be so shaped that the water will all drain to the air-pump.

**The Position of the Jet Pipe** or nozzle, and the form of the delivery openings in it depend so much on the form of the condenser that no definite rule can be given; the nozzle pipe may have a number of small holes drilled in it, or the water may issue from transverse or longitudinal saw-cuts or slots, or may all issue in a sheet, from the end of the pipe, through a nozzle resembling a section of pipe which has been flattened and nearly closed.

**The Quantity of Injection Water** depends on the weight of steam to be condensed, and on its heat, and,—to ascertain the exact quantity,—the temperature of the injection water, and the required temperature of hot-well must be known.

The vacuum with this type of condenser rarely exceeds 25 inches, and is more commonly 24 inches,—which corresponds to a temperature of about 140°; the temperature of the hot-well varies, in practice, from 110° to 130°.

The number of pounds of injection water (Q) necessary to condense one pound of steam is given by,—

**Rule 123.** $$Q = \frac{1114 + (\cdot 3 \times T_S) - T_H}{T_H - T_W},$$

where $T_S$ = temperature of steam at exhaust.*
$T_H$ = ,, of hot-well.
$T_W$ = ,, of cooling water.

**Rule 124.** It is usual to make an allowance of injection water of from 27 to 30 times the weight of steam to be condensed, for vessels in temperate climates, to 30 to 35 times for the tropics.

The relation of weight of steam to volume, to temperature, and to release pressure is given in the following Table :—

## Table LVI.—Weight, Pressure, and Temperature of Steam of Low Pressure.

| Pressure in | | Temperature F°. | Latent Heat F°. | Volume of one Pound of Steam. | Pressure in | | Temperature. F°. | Latent Heat F°. | Volume of one Pound of Steam. |
|---|---|---|---|---|---|---|---|---|---|
| Pounds Absolute. | Vacuum Ins. | | | cub. ft. | Pounds Absolute. | Vacuum Ins. | | | cub. ft. |
| 0·3 | 29·4 | 67·5 | 1070 | 1067 | 1·7 | 26·6 | 120·3 | 1029 | 200 |
| 0·4 | 29·2 | 74·0 | 1063 | 800 | 1·8 | 26·4 | 122·4 | 1028 | 190 |
| 0·5 | 29·0 | 80·0 | 1058 | 640 | 1·9 | 26·2 | 124·6 | 1026 | 181 |
| 0·6 | 28·8 | 85·5 | 1054 | 535 | 2·0 | 26·0 | 126·7 | 1025 | 172 |
| 0·7 | 28·6 | 90·4 | 1051 | 461 | 2·1 | 25·8 | 128·6 | 1024 | 165 |
| 0·8 | 28·4 | 94·5 | 1048 | 410 | 2·2 | 25·6 | 130·4 | 1022 | 158 |
| 0·9 | 28·2 | 98·5 | 1045 | 367 | 2·3 | 25·4 | 132·2 | 1021 | 152 |
| 1·0 | 28·0 | 102·0 | 1042 | 333 | 2·4 | 25·2 | 134·0 | 1020 | 145 |
| 1·1 | 27·8 | 105·0 | 1040 | 306 | 2·5 | 25·0 | 135·6 | 1019 | 140 |
| 1·2 | 27·6 | 108·0 | 1038 | 282 | 2·6 | 24·8 | 136·9 | 1018 | 135 |
| 1·3 | 27·4 | 111·0 | 1036 | 260 | 2·7 | 24·6 | 138·2 | 1017 | 130 |
| 1·4 | 27·2 | 113·7 | 1034 | 240 | 2·8 | 24·4 | 139·6 | 1016 | 125 |
| 1·5 | 27·0 | 116·0 | 1033 | 225 | 2·9 | 24·2 | 141·0 | 1015 | 121 |
| 1·6 | 26·8 | 118·2 | 1031 | 212 | 3·0 | 24·0 | 142·2 | 1014 | 118 |

**The Area of Injection Orifice,** and size of pipes, is governed by the head of water, the vacuum, and the length of piping and number of bends, &c., or, in other words, by the equivalent head at the condenser.

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* Strictly, *absolute* temperatures (Fahrenheit temperatures + 461°) should be used in this and all similar calculations.

Neglecting the resistance to flow at the orifice, and in the pipes and passages, the velocity at the condenser may be found as follows :—

Let $h$ be the head, in feet, above the valve on the condenser ; $p$, the pressure in the condenser in lbs. per sq. inch ; $h_1$ the equivalent head ; and $g$, gravity. Then,—

$$h_1 = h + (15 - p)\ 2{\cdot}3,$$

and velocity in feet per second is $\sqrt{2\ gh_1}$ or $8\sqrt{h_1}$.

**Rule 125.** In practice, owing to the loss of head resulting from resistances at valves, and in pipes, &c., the actual velocity is only about half that given by the above rule ; hence, in designing, it is usual to calculate on a velocity of only 25 feet per second for shallow draught steamers, and 30 feet per second for deeper ones.

The following more concise expressions are derived from the above rules,—

**Rule 126.**

$$\left.\begin{array}{l}\textbf{Area of orifice}\\ \textbf{in sq. ins.}\quad .\end{array}\right\} = \frac{\text{No. of c. ft. of injection water per minute}}{10{\cdot}4 \text{ to } 12{\cdot}5 \text{ according to circumstances}}.$$

**Rule 126a.**

$$\left.\begin{array}{l}\textbf{or Area of ori-}\\ \textbf{fice in sq. ins.}\end{array}\right\} = \frac{\text{Weight of injection water in lbs. per min.}}{650 \text{ to } 780 \text{ according to circumstances}}.$$

A snifting or overflow valve, held on its seat by atmospheric pressure only, should be provided, to prevent pressure or an undue accumulation of water in the condenser: its diameter should be the same as that of the injection valve.

## 2. Surface Condensers.

These are now supplied to all classes of vessels both for economy in working and to obtain the high vacuum so desirable when the maximum power is required from an engine, and so necessary to the turbine for high efficiency as well as maximum output. The saving in fuel alone as against the common condenser averages 15 per cent., and may be as much as 20.

The benefit of high vacuum is shown by the relative steam consumptions per horse-power, both with turbines and reciprocators, determined by the experiments of Sir C. Parsons and Professor Weighton.

**Table LVII.—Effect of Vacuum on Steam Consumption in lbs. per I.H.P. in a Turbine of 1000 H.P., and in a Quadruple and a Triple Expansion Experimental Reciprocator of 200 I.H.P.**

| Vacuum . . . Ins. | 20 | 22 | 24 | 26 | 27 | 28 | 29 |
|---|---|---|---|---|---|---|---|
| A turbine | 19·2 | 18·1 | 16·9 | 15·6 | 14·8 | 13·9 | 13·0 |
| A quadruple reciprocator | 16·7 | 16·1 | 15·5 | 15·0 | 14·7 | 14·5 | 14·3 |
| A triple ,, | 14·8 | 14·35 | 14·05 | 13·90 | 13·8 | 13·77 | 13·76 |

The surface condenser is itself more costly than a common one, and requires a separate air pump and water pump against the single pump combining the services with a jet condenser. The extra cost, however, is quite trifling compared with the advantages.

**The form of a Surface Condenser** is not of great consequence, but it is necessary that it shall be such that the economy of cost of construction does not reduce its efficiency, nor should it be that to gain a small increase in efficiency the cost is greatly increased. In a general way the cylindrical body is convenient, cheap to manufacture, and capable of an arrangement of tubes and baffles, whereby a high efficiency is attained in practical working. It is very light, requires no stays to stiffen it, and can be fixed in any convenient place. Mr Weir's condenser, known as the *Unaflow*, is heart-shaped or approximating to a spherical triangle in cross-section, generally placed vertically with the apex at the bottom, so that the steam expands over a large surface on entering the upper part and is caused to pass over the whole of the tubes below so that every square foot is active. The condensed water drops to the sides and drains to the bottom.

Mr Morison's contraflow system effects the same end in a body of different shape; in his condenser there is the same tapering of the body of tubes, and the steam is forced to pass over the whole, the water, however, is collected and led away to the air pump by special means provided. Mr Morison adopts a similar form of barrel, but more like the section of an egg than that of Mr Weir.

**Cooling Surface** is now entirely of brass tube, and the quantity depends on the requirements of each particular case. If high vacuum is desired the surface must be somewhat more than if a lower amount will do; but the temperature of the cooling water is the most serious factor, especially for high vacuum. Sea-water in several parts of the tropics is 85° F. in summer time, and 80° F. is common. In the North Atlantic generally and about the British coasts 60° F. is the usual summer temperature, and 65° F. a maximum. The temperature of steam or vapour at a pressure of 1 lb. is no more than 102° F., so that with a good supply of cooling water 28 inches of vacuum can be maintained, but 29 inches is impossible in the tropics, because the temperature of vapour is then only 80° F.

## Table LVIII.—Summer Temperature of Sea-Water at Various Parts of the World.

| Locality. | Temperature. | Locality. | Temperature. |
|---|---|---|---|
| | F°. | | F°. |
| North Sea, off Drontheim, | 60 | Indian Ocean, Kurrachi, | 85 |
| ,, ,, Bergen, | 63 | ,, near Colombo, | 85 |
| ,, ,, Gothenburg, | 61 | ,, ,, Bombay, | 84 |
| Baltic Sea, ,, Stettin, | 64 | ,, ,, Singapore, | 86 |
| ,, ,, Stockholm, | 50 | ,, ,, Penang, | 87 |
| ,, ,, Kronstadt, | 66 | China Seas, near Hong Kong, | 83 |
| Thames, Gravesend, | 64 | ,, ,, Woosung, | 69 |
| Scheldt, Antwerp, | 66 | ,, ,, Shanghai, | 64 |
| English Channel, | 62 | ,, ,, Kobe, | 67 |
| Portuguese Coast, | 72 | Atlantic Ocean, off Lisbon, | 72 |
| Mediterranean, Gibraltar, | 72 | ,, ,, Madeira, | 76 |
| ,, Marseilles, | 70 | ,, ,, Azores, | 76 |
| ,, Malta, | 75 | ,, ,, Cape Verde | 81 |
| ,, Port Said, | 76 | ,, North, in places, | 73 |
| Black Sea, near Bosphorus, | 74 | ,, off Ireland, | 66 |
| Red Sea, ,, Suez, | 87 | ,, ,, Boston, U.S.A. | 65 |
| ,, ,, Middle, | 90 | Gulf of Mexico, W. Indies, | 83 |
| ,, ,, Aden, | 87 | S. America, Buenos Aires, | 71 |
| Indian Ocean, West, | 85 | | |

*The Cooling Surface* should in a general way be in proportion to the weight of steam to be condensed. Professor Weighton with a contraflow condenser deposited 33 lbs. per sq. ft. per hour, and under very favourable conditions as much as 40 lbs. has been got. In Destroyers on trial, when new and the surface quite clean, 28 to 30 lbs. can be condensed. Mr Weir has been able to get 35 lbs. with a vacuum of 27·3 ins. Allowing for the surface losing efficiency after service, it may be taken that a square foot can be relied on to condense 20 lbs. with a liberal amount of cooling water. If the steam consumption of a triple engine be taken at 16 lbs. per I.H.P., then : (Rule 126) surface per I.H.P. $=\frac{16}{20}=0{\cdot}8$ sq. ft. In the same way for a turbine with a consumption of 13 lbs. the allowance should be : (Rule 126a) $\frac{13}{20}$ or 0·65 sq. ft. The following table may be taken as good practice under the conditions named. It may be said, however, that the ordinary cargo steamer with reciprocators, going to all parts of the world, has 1 sq. ft. of cooling surface per I.H.P., and this is sufficient for a good vacuum anywhere.

### Table LIX.—Cooling Surface. Allowance per I.H.P.

| Description of Engines. | Home Waters. | Tropics. |
|---|---|---|
| Triple Compound, Express Steamers, . | 0·80 sq. ft. | 1·23 sq. ft. |
| ,, ,, Economic Type, . | 0·70 ,, | 1·06 ,, |
| Quadruple Compound, Economic Type, | 0·65 ,, | 0·95 ,, |
| Turbine driven, Ordinary Type, . . | 0·65 ,, | 1·10 ,, |
| ,, ,, Express Type, . . | 0·80 ,, | 1·25 ,, |

In Naval ships; where full speed is only attained at intervals for a short time and the condensers of which are always clean, an allowance of half a square foot suffices for Destroyers and other very light high-speed craft, and for the ordinary cruiser and battleship with turbines 0·75 sq. ft. S.H.P. at full power. There is, however, a limitation to the condensing capacity of cooling surface if the supply of water is restricted. The total length of tube through which the water flows should not exceed 400 diameters in the temperate zone for high surface efficiency: 20 lbs. can then be relied on as the rate of condensation, but with so long a flow there will be a serious reduction in the tropics. If, therefore, high vacuum and good rate of condensation is required, with the cooling water 75° to 80° F., the traverse should not be more than is given by the following rule:—

**Rule 127.** Traverse of cooling water $= 400 \times \frac{6}{Q} \times \frac{d}{12}$,

Q is the quantity of water required to be condensed per square foot of surface per hour, $d$ is the external diameter of tube in inches.

The traverse should not be more than 330 diameters of tube for ships visiting the tropics and requiring high vacuum.

It was usual for the cooling water to travel three times through the tubes; now, with tubes of considerable length, twice through suffices. The flow over the cooling surface should be rapid for good results, and is often at the rate of 300 ft. per minute, this, however, is high for small tubes and requires large power to force it through; 200 ft. per minute is a good rate. The coldest water must be applied to the coolest part of the condenser, so as to maintain the efficiency of the surface as uniform as possible and to get the best results from the whole; hence the cooling water should be admitted at the bottom of the condenser and discharged from the top, where the difference in temperature from the incoming steam is not great but still sufficient to condense a portion. The difference in temperature at discharge for economic working will be about 5° F., and in the tropics even less.

The quantity of cooling water can be estimated by the following formula:—

**Rule 128.** $Q = (T_1 + L) - T_3 \div (T_2 - T_0) = \dfrac{1114 + 0{\cdot}3\ T_1 - T_3}{T_2 - T_0}$,

$T_1$ is the temperature of the steam entering the condenser, and
L its latent heat;
$T_0$ the temperature of the cooling water at entry, and
Q the quantity of the cooling water at entry;
$T_2$ the temperature of cooling water on leaving the condenser,
$T_3$ the temperature of the hot-well.

The following table gives the ratio of least cooling water to the steam condensed at different temperatures of the sea.

### Table LX.—Ratio of Cooling Water to Steam Condensed.

| Vacuum . Ins. | 25·0 | 25·5 | 26·0 | 26·5 | 27·0 | 27·5 | 28·0 | 28·5 | 29·0 | 29·3 |
|---|---|---|---|---|---|---|---|---|---|---|
| Sea water, 50° F. | 13·9 | 14·7 | 15·4 | 16·4 | 18·2 | 20·1 | 22·8 | 28·0 | 40·4 | 65·0 |
| ,, 60° F. | 16·0 | 17·1 | 18·1 | 19·5 | 22·1 | 24·8 | 29·0 | 38·0 | 64·3 | 158 |
| ,, 70° F. | 18·9 | 20·5 | 21·8 | 24·0 | 27·9 | 32·3 | 40·0 | 66·3 | 156 | ... |
| ,, 80° F. | 23·1 | 25·5 | 27·6 | 31·0 | 37·9 | 46·3 | 63·0 | 133 | ... | ... |
| ,, 85° F. | 26·0 | 29·1 | 32·8 | 40·0 | 50·5 | 61·5 | 153 | ... | ... | ... |

The table shows the least weight of cooling water required in practice per pound of steam entering the condenser at a pressure of 12 lbs. absolute. Modern air-pumps can maintain a vacuum of 29 inches, but it will be seen by this table that the quantity of cooling water required for it at a temperature of 70° is enormous, and with water at 80° is impossible.

The highest possible vacuum with the temperature of cooling water at 80° is 28·8; and the ratio no less than 270. The highest vacuum when the water is 70° will be 29·1 inches, and the ratio then 220. In the temperate zone, with sea-water at 60°, a vacuum of 29·4 can be maintained by passing 280 times the weight of steam condensed.

In every-day practice, and the condenser not too clean, the quantity of water required may easily be 10 to 20 per cent. more than that given in this table.

**Condenser Tubes.**—These are, as a rule, made of brass, solid drawn, the composition being 70 per cent. of best selected copper, and 30 per cent. of best Silesian spelter; the Admiralty, however, specify 70 per cent. copper, 29 per cent. spelter, and 1 per cent. tin.

Condenser tubes vary in diameter from about 9/16-inch in Destroyers to about 1 inch in large merchant steamers, but 3/4-inch tube, with a thickness of 18 L.S.G. (·048 in.), is most commonly used in the Merchant Service, and 5/8-inch tube of 19 L.S.G. in the Navy.

**Length unsupported.**—(Rule 129.) When tubes are secured at the tube-plates by screw ferrules and packing, the unsupported length should not exceed 100 diameters; or, when they are held by tightly fitting ferrules, 120 diameters. If the tubes are longer than this, they should be supported by intermediate diaphragms of rolled brass, from ⅜-inch to ½-inch in thickness, according to size; but care must be taken, in arranging the design, that these diaphragm plates do not interfere with the free flow of the steam to all parts of the condenser.

**Tube plates.**—These should be of rolled brass of the following thicknesses:—

**Rule 130.**

**When wood ferrules are used . . 1·5 × diameter of tube.**

**" screw glands " . . 1·1 × " "**

"Diameter of tube" signifies, in all cases, *external* diameter.

Fig. 24 shows the method of securing a tube by a wood ferrule, and fig. 25 by means of a screw gland and packing; the dimensions on the figures give the proportions ordinarily adopted for ¾-inch tubes.

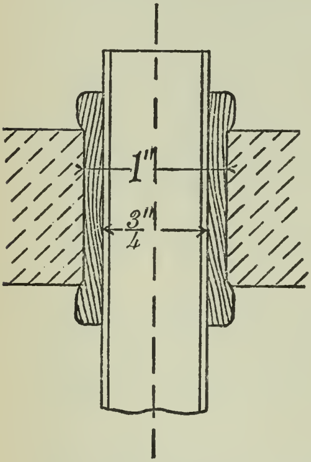

FIG. 24.

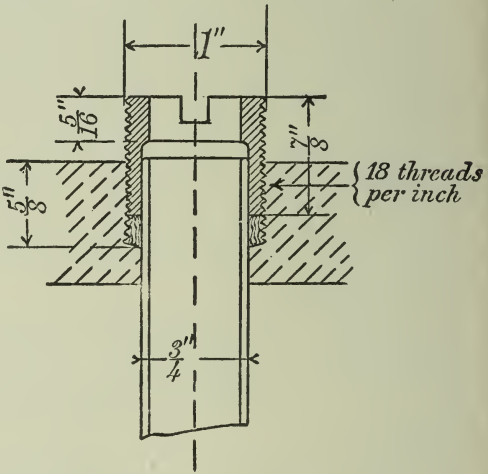

FIG. 25.

**Spacing of Tubes.**—The tubes should be spaced zigzag, and equilaterally (*i.e.* the centre of any tube should be at the same distance from the centre of each of the immediately surrounding tubes). The pitch may be as follows:—

**Rule 131.**

**When wood ferrules are used—Dia. of hole for ferrule + ¼ in.**

**When screw glands are used—Extreme dia. of gland + ⅛ to $^3/_{16}$ in.**

The following Table will be of assistance in making rough calculations for size of condenser:—

## Table LXI.—Size of Surface Condenser.

| External diameter of tube—¾ inch. | | | | | |
|---|---|---|---|---|---|
| Pitch of tubes, . . | $1\frac{1}{8}''$ | $1\frac{5}{32}''$ | $1\frac{3}{16}''$ | $1\frac{7}{32}''$ | $1\frac{1}{4}''$ |
| Number of tubes per sq. ft. of tube plate, . . . . | 136 | 128 | 122 | 116 | 110 |
| Cooling surface per cubic ft. of tubes' space, . . . . | 26·7 sq. ft. | 25·1 sq. ft. | 23·9 sq. ft. | 22·7 sq. ft. | 21·5 sq. ft. |

The surface of 1 foot of ¾-inch tube is ·1963 sq. ft., and of ⅝-inch is ·1637 sq. ft.

See also Table of Surfaces of Tubes, Table CLXI.

**Application of Cooling Water.**—The simplest method of applying the cooling water is to allow it to run direct from the sea into the condenser, and then to pump it overboard from the condenser; where this cannot be conveniently arranged, and the water is forced through the condenser by a reciprocating pump, a large air-vessel should be provided as near to the pump as possible, to guard against shock from "racing" or sudden starting.

The centrifugal pumps, invariably employed for circulating purposes on ships of war and in passenger steamers and in all ships of large power, suck from the sea and drive the water through the condensers, but they are of course incapable of communicating any shock to the structure.

In order to reduce the shock from reciprocating circulating pumps, some engineers fit a screw-down by-pass valve which admits air when opened more or less, whenever "racing" occurs.

In merchant steamers the condenser was commonly a part of the framing, and made of cast iron; its strength of course depended largely on its form and position, and had little relation to the stresses resulting from its action as condenser.

In war vessels, on the other hand, the condenser never forms any part of the framing; it was commonly cylindrical in form, made of naval brass or gun-metal, and strong enough to bear an internal test pressure of 30 lbs. per sq. inch. It is now in ships generally of sheet steel, heart-shaped in transverse section and having cast-iron waterways and covers at the ends.

Condensers for Destroyers, &c., up to about 5 feet diameter, usually had the barrels made of sheet brass (8 to 6 L.S.G.), with flanges and stiffening rings riveted and soldered on.

Manholes, sight-holes, mud-holes, and air-cocks should always be provided; in Naval condensers fittings for introducing soda, and for boiling out, are always fitted in addition.

## AIR-PUMPS.

Wherever practicable these should be vertical, and single-acting, since this type is in every way the most efficient and satisfactory. Centrifugal pumps are now specially designed and act well as air-pumps in conjunction with ejectors for producing vacuum. *The Kinetic Air-Pump System* is now taking the place of reciprocating pumps for independent condensers for auxiliary machinery.

In vertical screw engines the air-pump should be driven by levers and links from the L.P. crosshead or gudgeon; when driven direct from the piston or crosshead the speed is objectionably high, and the pump is not usually convenient of access. On the other hand, the clearance space is relatively small and it has no gear or parts requiring to be lubricated.

In twin-screw engine installations when the screws turn outwards the air-pump is in front of the engine, worked by levers in the usual way, the condenser being near the ship's side. When the screws turn inwards the pump is at the back, as with single-screw engines. With turbines the condensers are often beside the L.P. member, and the air-pumps being driven by independent motors are placed in any suitable place contiguous.

In paddle-engines the air-pumps are placed in different positions, and driven in different ways according to the type of engine. In oscillating engines they are usually inclined, and driven by connecting-rods from a crank formed in the middle of the intermediate shaft, or by an eccentric fixed in the same position; but are sometimes placed vertically, and driven through a lever or beam. In almost all other types of paddle engine they can be placed vertically without difficulty, and may be driven as found most convenient.

**Size of Air-Pump.**—Whether for jet or surface condenser, the quantity of water to be removed by the air-pump can be easily determined by the rules given under "Condenser," but the quantity of air is variable and cannot be calculated; and it is therefore necessary, especially in the case of surface condensing engines, to rely more on experience than on theoretical considerations.

**The Capacity of Air-Pump** depends on the amount of water passing through it from the condenser, inasmuch as the air to be abstracted will vary with the amount of feed water. If a condenser is absolutely tight, the engine glands leak no air, and the circuit of the feed water is closed so that no fresh air comes in contact with it, a very small air-pump suffices to maintain a good vacuum when once formed. But these are conditions not to be counted on in practice; moreover, it is necessary, especially in some services, to have a good vacuum very quickly after the pump starts. The height of the vacuum, or the rarefaction of the condenser, does not depend on the size of the air-pump, but on its fitness; for an ill-designed pump, however large, cannot produce high vacuum, and most pumps act efficaciously from all clearance spaces being filled and air leaks stopped by the water passing through them; without water they would fail to make or maintain a good vacuum. Under fair working conditions, such as obtain on shipboard, a vacuum of 26 ins. can be maintained with an air-pump having a capacity of 0·3 cubic foot for each pound of condensed water passing through it. Hence for a triple engine using 16 lbs. of steam per H.P. hour at R revolutions,

$$\text{the amount passing through the pump each stroke} = \frac{16 \times \text{I.H.P.}}{60 \times \text{R}}.$$

$$\text{Then capacity of pump} = \frac{0{\cdot}3 \times 16}{60} \times \frac{\text{I.H.P.}}{\text{R}} = 0{\cdot}08 \frac{\text{I.H.P.}}{\text{R}} \text{ cub. ft.}$$

This, however, would be too small for quick formation of vacuum, and with leaky L.P. glands insufficient to keep the 26 ins. To maintain as well as form a vacuum of 28 to 29 ins. the allowance should be 0·8 cubic foot per pound of feed water. Then

**Rule 132. Capacity of air-pump** $= \frac{0{\cdot}8 \times f}{60} \times \frac{\text{I.H.P.}}{\text{R}}$ when worked by the engine $= 0{\cdot}0133 f \times \frac{\text{I.H.P.}}{\text{R}}$.

For triple compound engines $f$ may be taken at 16.
,, quadruple ,, ,, 14.5.

Taking 15 as the average amount, the general rule becomes

$$\text{Capacity of air-pump} = 0{\cdot}2 \times \frac{\text{I.H.P.}}{\text{R}}.$$

It is, however, often convenient in designing and common practice to make the air-pump in proportion to the L.P. cylinder. Hence:—

**Rule 132a. Capacity of Single Acting Air-Pump** = 0·04 that of the L.P. cylinder.

For ferry steamers or those calling at piers, and so often stopping and required to get away at full speed rapidly, the multiplier may be 0·047 with advantage.

The common air-pump with foot, bucket, and head valves is not often used now, the *Edwards* pump with only head valves having superseded them, as being quite as efficient in every-day practice, and less costly both to make and to work by the omission of two-thirds of the valves, and less likely to get out of order also.

**Weir's dual Air-Pumps** when worked by an independent motor are also much in favour, especially for turbines. Here the work is done in stages, the one pump delivering semi-compressed air into the other and keeping all its air leaks sealed and its temperature low by drawing and using some specially cooled water. The delivering pump may be warmer and not so carefully sealed.

**Parsons' Vacuum Augmenter** is an ingenious and simple method of getting high vacuum. In this the air-pump is placed well below the condenser and permits of a column of water to seal the air, which is drawn away by a steam ejector from the condenser bottom, and keeps the condenser itself free from air or any other gaseous matter, which would otherwise seriously reduce its efficiency.

**Air-Pump Rods.**—These are made of one of the rolled strong bronzes. Where a guide is fitted, and the stroke of the pump does not exceed 2¼ times its diameter, the rod (without regard to its material) may have a diameter given by,—

**Rule 133.** $$\text{Diameter of rod} = \frac{\text{Diameter of pump}}{9} + \cdot 6 \text{ inch.}$$

See Table LXII. (page 163) for sizes given by this rule.

**The Size of the Screw** on the air-pump rod, by which it is attached to the air-pump crosshead, is given by,—

**Rule 133a.** $$\text{Area at bottom of thread} = \frac{\text{Area of bucket}}{20 \times \sqrt{\text{diameter of bucket}}}.$$

See Table LXII. (page 163) for sizes given by this rule.

This allows a stress of about 2000 lbs. per sq. inch of material in the rod for a 12-inch pump, increasing to 3800 lbs. per sq. inch in the rod for a 40-inch pump, the load on the bucket being taken as 30 lbs. per sq. inch.

When the number of reciprocations is high (say 200 to 350 per minute) the rods should be made 10 per cent. larger in diameter than given by the rule, although the stroke may be short, or even very short, since there is always a liability to sudden strain, owing to irregular or intermittent action of the pump, and the intensity of these strains will vary as the squares of the bucket speeds,—other things being equal.

**Connecting-Rods for Trunk Air-Pumps.**—When of the type commonly employed for driving the pumps of oscillating paddle-engines these may have the following proportions :—

**Rule 134. Area of section of rod = ·01 × area of bucket ;**
or, in case of a round rod,—

**Rule 134a. Diameter of rod = ·1 × diameter of bucket ;**
when two bolts are used to connect the brasses at the end,—

**Rule 134b.**

**Diameter of each bolt (in body) = ·056 × diameter of bucket.**

Air-pump valves are usually of metal, and made as light as possible with little lift. Simple discs of sheet brass answer very well ; if fitted as tongues it is necessary to employ a metal with some spring and capable of withstanding continuous vibratory stresses. Sheet phosphor-bronze is good.

When the air-pump valves are of india-rubber care should be taken to fit a quality that is capable of resisting the heat, &c.; the Admiralty specification is as follows:—

"To be made of the best Para caoutchouc, with no other materials whatever than sulphur and white oxide of zinc; the sulphur is not to exceed 1¾ per cent., and the oxide of zinc not to exceed 70 per cent. The india-rubber is to be made of the best materials, not re-manufactured, and to be of a homogeneous character throughout, thoroughly compressed, free from air-holes, pores, and all other imperfections.

"Samples of the india-rubber will be tested, and must be capable of enduring a dry heat test of 270° Fahrenheit for one hour, and a moist heat of 320° Fahrenheit for three hours, without impairing its quality."

**Air-Pump Barrels.**—In the mercantile marine the pump barrel is of bronze, secured to its seat and supporting the delivery chamber and hot-well by flanges. It is usual in Naval work to make barrel, head and foot boxes all one bronze casting. Whichever type of construction is used, the working barrel should be of thickness given by,—

**Rule 135.**

$$\text{Thickness of air-pump barrel} = \frac{\text{Diameter of pump}}{60} + \cdot 25\text{-inch.}$$

See Table LXII. (page 163) for thicknesses given by this rule.

When the pump is of the Naval type the portions above and below the working barrel may be from $\frac{1}{16}$ to ⅛ inch thinner than the barrel, according to size and design.

When the bronze barrel is not enclosed in an outer casting, and there are foot valves and bucket valves fitted, it is very desirable to have a manhole in the side, through which access may be had to them without disturbing the cover or guides. Similarly, when the size of the pump renders it possible, hand or sight holes should be provided in the sides of the head box, for getting at the head valves.

**The Areas through the Valve Gratings** of foot, bucket, and head valves should be kept as large as possible; on an average, those of the foot and bucket valves run about one-third of the gross area of bucket, and those of the head valve a little larger; they can of course be increased by increasing the diameter of the pump, and decreasing the stroke.

Edwards' pumps having only "head" valves are now almost always used for air-pumps, and give every satisfaction, as they can maintain high vacuum without foot or bucket valves to trouble one, and with an unpacked bucket working with the minimum amount of friction.

**Speeds of Bucket.**—In ordinary mercantile engines the speed of bucket employed varies from 200 feet to 350 feet per minute; but in Naval engines it commonly ranges from 300 feet to 450 feet, and with pumps driven direct from the pistons is of course higher still. When the pump works in connection with a jet condenser the speed should not much exceed 200 feet per minute.

**Sizes of Suction and Discharge Pipes.**—For the pumps of surface condensing engines (where no jet apparatus is fitted) the diameter of suction pipe from condenser, in inches, is given by,—

**Rule 136. Diamr. of air-pump suction pipe in ins.** $=\frac{D}{48}\sqrt{S \times R}$,

or, **Equivalent mean speed in pipe must not exceed 2500 ft. per min.**, where D is diameter of bucket in inches, and S its stroke in feet, R the revolutions per minute.

If there is a supplementary hot-well or feed-tank the pipe connecting it with the air-pump should be of diameter given by,—

**Rule 136a. Diamr. of air-pump delivery pipe in ins.** $=\frac{D}{33}\sqrt{S \times R}$.

In addition, an air-pipe should be fitted as close to the pump as possible; its diameter may be,—

**Rule 137. Diameter of air-pipe in ins.** $=\frac{D}{100}\sqrt{S \times R}$,*

or, **Equivalent mean speed in pipe must not exceed 6400 feet per minute.**

If the pipe to feed-tank is long, or has many sharp bends in it, these points must, of course, be considered and allowed for in fixing the size.

When the pump works in connection with a jet condenser, the diameter of suction and delivery pipes may be,—

**Rule 138. Diamr. of suction and delivery pipes in ins.** $=\frac{D}{25}\sqrt{S \times R}$,

or, **Equivalent mean speed in pipe must not exceed 625 feet per minute.**

The delivery pipes from air-pumps of jet-condensing engines should have good large air-vessels (say capacity not less than capacity of pump) placed as near to the pumps as possible.

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* *Note.*—Where there is only one delivery pipe from pump head, its diameter must be $\frac{D}{48}\sqrt{S \times R}$ ins.

Table LXII.—Air and Circulating Pumps.

| Diameter of pump (single-acting). 1 | Thickness of barrel (single-acting). 2 | Thickness of barrel +10 per cent. (double-acting). 3 | Diameter of rod. 4 | Diameter of screw on outer end of rod. 5 | Diameter of double-acting pump of same capacity (circulating only). 6 |
|---|---|---|---|---|---|
| inches. | inch. | inch. | inches. | inches. | inches. |
| 8 | ⅜ | ... | 1½ | 1¼ | ... |
| 10 | 7/16 | ... | 1¾ | 1½ | ... |
| 12 | 7/16 | ... | 2 | 1¾ | ... |
| 14 | ½ | ... | 2¼ | 2 | ... |
| 16 | ½ | ½ | 2½ | 2¼ | 11½ |
| 18 | 9/16 | ½ | 2½ | 2¼ | 13 |
| 20 | 9/16 | 9/16 | 2¾ | 2½ | 14 |
| 22 | ⅝ | 9/16 | 3 | 2¾ | 15½ |
| 24 | ⅝ | 9/16 | 3¼ | 2¾ | 17 |
| 26 | 11/16 | ⅝ | 3½ | 3 | 18½ |
| 28 | 11/16 | ⅝ | 3¾ | 3 | 20 |
| 30 | ¾ | 11/16 | 4 | 3¼ | 21 |
| 32 | ¾ | 11/16 | 4¼ | 3½ | 22½ |
| 34 | 13/16 | 11/16 | 4½ | 3½ | 24 |
| 36 | ⅞ | ¾ | 4½ | 3¾ | 25½ |
| 38 | ⅞ | ¾ | 4¾ | 4 | 27 |
| 40 | ⅞ | 13/16 | 5 | 4 | 28½ |

## RECIPROCATING CIRCULATING PUMPS.

These may be double-acting, except in small ships when they are single-acting, and, as their efficiency is practically unaffected by position, they may be either vertical, horizontal, or inclined. In general design the various parts may resemble the corresponding parts of the air-pump (in practice the same patterns are sometimes used for both pumps), but circulating pumps require, in addition, large air vessels, non-return air or pet cocks, and, in the case of large double-acting pumps, a by-pass valve.

The air-vessel for a single-acting pump should have a capacity equal to twice that of the pump, when possible, but never less than one and a half.

The by-pass valve may have an area of about one-tenth the area of the bucket.

When the size of pump required would, if made single-acting, exceed 20 inches in diameter, it is better to fit a double-acting pump of half the bucket area, and thus obtain a steadier delivery, whilst at the same time reducing the magnitude of the stresses and the sizes of the various parts.

**Size of Circulating Pump.**—This depends on quantity of cooling water required, and number of strokes per minute.

Let $Q$ = cubic feet of cooling water required per minute.
$n$ = number of strokes per minute.
$S$ = length of stroke in feet.

Then,—

**Rule 139.** **Capacity of Circulating Pump** $= \frac{Q}{n}$ cubic feet.

and,—

**Rule 139a.** **Diameter of Circulating Pump** $= 13{\cdot}55\sqrt{\frac{Q}{n \times S}}$ inches.

In determining Q, the temperature of cooling water should be taken at the highest ordinary temperature encountered on the routes the vessel is intended to steam over, *v.* Table LVIII., page 153.

Q is the quantity of cooling water in cubic feet, $n$ the number of strokes per minute, S the stroke in feet.

If the allowance is 42 times the feed, and feed is taken at 15 lbs. per H.P., then,—

$$Q = \frac{42 \times 15}{62{\cdot}5 \times 60} \times \text{I.H.P.} = 0{\cdot}17 \times \text{I.H.P. cubic feet.}$$

**Rule 140.** **Then Capacity of Pump** $= \frac{0{\cdot}17 \times \text{I.H.P.}}{n}$ cubic feet,

if the pump is single-acting, $n$ = Revs. per min.

**Circulating Pump Rods.**—For single-acting pumps, these should be of the same materials as the air-pump rods, and the same formula may be used to obtain size. See Table LXII.

Wherever possible the air and circulating pump rods should be made of exactly the same dimensions, in order that one spare rod may serve for either pump.

The rule for rods of double-acting circulating pumps will then be,—

**Rule 141.** **Diameter of Rod** $= \frac{D}{6{\cdot}4} + {\cdot}6$ inch (see Table LXII.).

Although packing is occasionally used, the buckets of circulating pumps really do not require any, and are better without it, since the serious friction is thereby saved.

The thicknesses of gun-metal barrels or liners for circulating pumps may be determined by the rule previously given for air-pump barrels (Rule 135), when the barrels are of similar type and similarly supported. When the barrel is fixed in place by means of a single flange placed at mid-length (a common method for double-acting circulating pumps), the thickness should be 10 per cent. above that given by the rule (see Table LXII.).

The areas through the foot, bucket, and delivery valves should be kept as large as possible, but, as in the case of air-pumps, they will usually average one-third of the gross bucket area.

The bucket speeds in common use are from 200 to 400 feet per minute.

The sizes of suction and delivery valves may be as follows:—

**Rule 142.** **Diameter of Pipe** $=\frac{D}{F}\times\sqrt{S.}$ ins.

Where $D$ is the diameter of bucket in inches, $S$ its mean speed in feet per minute, and $F$ a co-efficient, the values of which are as follows:—

| | | | | |
|---|---|---|---|---|
| **Double-acting pumps** | **Suction** | —$F=25$ | (mean speed in pipe, | 625 feet). |
| | **Delivery** | —$F=24$ | ( ,, ,, | 576 ,, ). |
| **Single-acting pumps** | **Suction** | —$F=27$ | ( ,, ,, | 729 ,, ). |
| | **Delivery** | —$F=24$ | ( ,, ,, | 576 ,, ). |

When any pipe is under 10 inches in diameter the velocity of flow through it should be kept 10 per cent. below that allowed by the above rule. Specially long or specially tortuous pipes must also have special consideration,—the velocity of flow being reduced in accordance with the circumstances of the case.

Lloyd's rules require that a bilge injection or a bilge suction to the circulating pump shall be fitted, the diameter of which shall be at least two-thirds that of the sea-inlet.

## CENTRIFUGAL CIRCULATING PUMPS.

In Naval and express work this type of pump is almost invariably used, and not infrequently fitted in other classes of mercantile steamers. It is, no doubt, preferable to the reciprocating type where large bodies of water are to be moved against a merely nominal head, and especially where the main engines run at a high number of revolutions per minute, since there is an entire absence of shock, and the work is done with a less expenditure of power. On the other hand, the centrifugal pump is more costly than the reciprocating, occupies more space, requires more attention.

**Size of Pipes.**—The size of pipes should be such that the speed of water does not exceed 500 feet per minute, even when the sea temperature is 75° ; or, in other words, when the sea temperature is 60° the speed should be about 330 feet per minute. When the pipes are large, say over 15 inches, the velocity of flow at full speed may be higher, so that in ships of very large power it is from 600 to 700 feet.

If the pipes are not short and direct, with easy bends, the efficiency of the pump will be improved by keeping the speeds 10 to 15 per cent. per min. lower.

A convenient form of the rule for size of pipe will then be as follows:—

When the cooling water is not more than 8 lbs. per H.P. minute or about 32 times the feed, the suction and delivery pipes may be as follows:—

**Rule 143.**

$$\text{Diameter of cooling water pipes in ins.} = \sqrt{\frac{\text{I.H.P.}}{25}} + \tfrac{1}{2}\text{ in.}$$

For ships going to the tropics and requiring a vacuum not less than 28 inches, then

$$\text{Diameter of cooling water pipes in ins.} = 0{\cdot}28\sqrt{\text{I.H.P.}} + \tfrac{1}{2}\text{ in.}$$

**Rule 143a.** $\text{Diameter of pipe in inches} = {\cdot}6\sqrt{Q} + \tfrac{3}{4}\text{ in.}$,

where Q is the maximum number of cubic feet of cooling water per minute.

**Size of Impeller or Wheel.**—A good proportion for the diameter of pump wheel is,—

**Rule 144.**

**Diameter of Pump wheel in inches = 2·8 × Diameter of pipes,**

or,—

**Rule 144a.** $\text{Diameter of pump wheel in inches} = 1{\cdot}8\sqrt{Q}$,

where Q is maximum number of cubic feet of cooling water per minute or, reversing the equation,—

**Rule 144b.** $Q = \dfrac{D^2}{3{\cdot}24}$ cubic feet.

**Width of Wheel at Periphery.**—The width of wheel at the periphery, or width of vane at the tip may be,—

**Rule 145.**

$$\text{Width of Wheel at Periphery} = \frac{\text{Diameter of Pipes}}{4}.$$

**Stroke of Piston.**—The stroke of the centrifugal pump engine may be conveniently fixed as follows:—

**Rule 146.** **Stroke of piston = ·3 × Diameter of impeller.**

**Size of Steam Cylinder.**—As centrifugal circulating pumps are invariably fitted with bilge suctions so as to be available as bilge pumps in case of need, and as the work done when pumping from the bilge is far in excess of that done when merely circulating water through the condenser, the size of steam cylinder must evidently be fixed with regard to the former duty.

The efficiency of the pump depends very greatly on the pipe arrangement; with direct pipes of good size, and easy bends, it may, when pumping from the bilge, approach 30 per cent., but a bad arrangement, with sharp bends, will very seriously affect it, and may easily reduce it below 20 per cent.; in ordinary practice, where reasonable care is exercised in scheming the pipes, it may be assumed to be 25 per cent. The efficiency also tends to diminish as the lift increases, so that a rather lower efficiency should be assumed in the case of large deep ships, and *vice versâ.*

$$\text{Then } \frac{\text{Indicated Horse-power}}{\text{Water Horse-power}} = 4$$

where "water horse-power" is $\frac{\text{lbs. water pumped per min.} \times \text{lift in ft.}}{33{,}000}$

or, putting it into another shape,—

**Rule 147.**

**Tons water pumped per hour × lift in feet × 150 = $P_m \times A \times S$,**

where $P_m$ = mean pressure in steam cylinder;
A = area of piston in square inches;
S = piston speed in feet per minute.

**Speed of Periphery of Pump Wheel.**—The speed of periphery of pump wheel necessary to discharge water at the velocities mentioned above against a head ($h$), under average conditions, is given by,—

**Rule 148.** $V = 11\sqrt{h}$,

where V is velocity of periphery in feet per second, and $h$ the actual lift in feet. With a very good arrangement of pipes the co-efficient may be as low as 9, in place of 11, but a bad arrangement will send it up to 13 or over.

In ordinary circulating work when inlet and outlet are both submerged, as is usual in Navy, V requires to be about 24 feet per second (1440 feet per minute), and the resistances may therefore be assumed to be equivalent to a head of nearly 5 feet.

When pumping from the bilge, V varies from 30 feet to 50 feet per second (1800 feet to 3000 feet per minute) in ordinary Naval work, but, as shown by the above equation, it increases as the square root of the head.

**Size of Bilge Suction Pipe.**—A convenient rule for area of this pipe is,—

**Rule 149.**

**One square inch area for each 7 tons per hour to be pumped.**

This limits the speed of water in the suction pipe to 580 feet per minute.

It is not desirable to have the area of bilge suction more than about ·6 of the area of the circulating pipes, as the size of steam cylinder required soon becomes cumbersome for the ordinary circulating work.

The pump should be capable of performing the specified bilge duty with a steam pressure not exceeding two-thirds of the ordinary boiler pressure.

## FEED PUMPS, &c.

**Capacity of Feed Pumps in Jet-Condensing Engines.**—This depends mainly on the degree of saltness at which the water in the boilers is to be maintained; or, in other words, on the proportion of the *gross* feed-water that is "blown off."

Let Q represent the *nett* feed-water, or the quantity required as steam by the engines, and say water in boilers is to be kept down to $n$ times the density (or saltness) of sea-water, then the amount of sea-water that must be pumped into boilers is,—

$$\textbf{Gross feed-water} = \frac{n}{n-1} \times Q.$$

To reduce the time required to bring the water in the boilers up to the working level again after "blowing off," it is usual to make *each* feed pump capable of pumping *twice* this quantity; therefore,—

**Rule 150.** **Each pump should supply** $\frac{2n}{n-1} \times Q.$

The **nett** feed-water (Q) required by a jet-condensing engine, working with steam of about 30 lbs. pressure, was 26 lbs. per I.H.P. per hour.

The amount of scale or salt deposited in the boilers does not depend on the density at which the water is kept, but only on the quantity of sea-water evaporated by the boilers.

The Hydrometer, or Salinometer, by means of which the density of the water is ascertained, is, in the Navy, graduated in degrees, so that, when floating in pure water, the zero point is at the surface, and when in clean sea-water it marks 10°, when in water of twice the density of sea-water, 20°, and so on. In the Merchant Service, engineers either express the density in ounces per gallon (sea-water containing about 5 oz. per gallon), or in "thirty-twos,"—sea-water containing about $1/32$ of its weight of solid matter.

**Capacity of Feed Pumps for Surface-condensing Engines.**—It will be sufficient if the total capacity of each pump be made equal to **twice** the nett feed-water required; this allowance also covering any want of efficiency in the pumps.

If Q be the nett feed-water required in lbs. per hour, $l$ length of stroke of feed pump in inches, and $n$ number of strokes per minute; then,—

**Rule 151.**

$$\textbf{Diameter of each feed-pump plunger, in inches} = \sqrt{\frac{1{\cdot}15 \times Q}{l \times n}}.$$

For ordinary compound engines the nett feed-water (Q) required may be taken at 18 lbs. per I.H.P. per hour, for triple engines 16·0 lbs. per I.H.P. per hour, and for quadruple engines 14·5 lbs. per I.H.P. per hour.

Or, if the formula be written,—

**Rule 152.** **Capacity of feed pump, in cubic inches** $= \frac{\text{I.H.P.}}{R} \times C$,

the values of C will be as follows for the nett feed:—

**For Compound engines,** $C = 8·3$.
,, **Triple engines,** $C = 7·4$.
,, **Quadruple engines,** $C = 6·7$.

For actual size of pump when worked from the main engine C should be, in each case, taken at double the value to allow for wastage and sudden demands for a larger than normal supply.

Where the feed pumps are driven by the main engines, there should be, except in very small engines, two pumps, each capable of supplying the boilers at full power, and so arranged that either may be worked independently of the other, and be easily put out of gear when not required.

**Relief Valves.**—When the pumps are driven by the main engines, each pump should be fitted with a relief valve, of a diameter equal to two-thirds that of the delivery pipe from that pump, and loaded to 1½ times the boiler pressure. All water from relief valves should go back into suctions. Independently driven feed pumps without fly wheels need no relief valves, as they simply stop automatically and go on again when the check valves on the boilers permit.

**Feed-pump valve-boxes and valves.**—These should be of best bronze; and since the seats, as well as the valves, wear out rapidly, loose ones should be fitted. The faces of the valve seats should be made flat, and the seating area, or area of faces in contact, should not be less than 20 per cent. of the gross area through valve; that is,—

**Rule 153.**

**Width of faces in contact** $= ·0475 \times$ **Diameter of valve.**

The diameter of valve, measured inside the seat, should be equal to that of the delivery pipe.

The noise made by the valves on their seats may be much reduced by loading them with light spiral springs, made of plated steel or of hard brass wire, and very large ones may have a cataract buffer.

**Feed pipes.**—The pipes leading to and from the feed pumps should be of such size that the velocity of flow, when working steadily at full power, does not exceed 500 feet per minute. The velocity should also be less as the pipes get smaller, to allow for the increased percentage of friction; and, if 400 feet be adopted as the limit in a 4-inch pipe, the limit in a $1\frac{1}{2}$-inch pipe should be 250 feet; or, more briefly, velocity for any diameter of pipe should not exceed that given by,—

**Rule 154.** $$V = 200\sqrt{\text{Diameter}},$$

where V is in feet per minute and diameter is in inches.

The maximum deliveries of pipes, on ordinary service, may be shown in tabular form as follows:—

### Table LXIII.—Delivery of Feed Pipes at Service Speeds.

| Diameter of pipe. | Delivery in cubic feet per minute. (C) | Delivery in lbs. per hour. | Diameter of pipe. | Delivery in cubic feet per minute. (C) | Delivery in lbs. per hour. |
|---|---|---|---|---|---|
| 1½ | 2·45 | 9,260 | 3½ | 20·43 | 77,200 |
| 1¾ | 3·60 | 13,600 | 3¾ | 24·30 | 91,800 |
| 2 | 5·04 | 19,050 | 4 | 28·63 | 108,200 |
| 2¼ | 6·79 | 25,660 | 4¼ | 33·30 | 125,800 |
| 2½ | 8·81 | 33,300 | 4½ | 38·42 | 145,200 |
| 2¾ | 11·22 | 42,400 | 5 | 50·00 | 189,000 |
| 3 | 13·94 | 52,700 | 5½ | 63·36 | 239,500 |
| 3¼ | 17·00 | 64,200 | 6 | 78·92 | 298,300 |

Then, to find diameter of feed pipe, determine value of C in equation,—

**Rule 155.** $$\frac{D^2 \times S}{184} = C,$$

where D is diameter of plunger in inches, and S its mean speed in feet per minute; look for the corresponding figure in column 2 or 5 of Table; and opposite, in column 1 or 4, will be found the appropriate diameter of pipe. If the exact figure does not appear in column 2 or 5, take the next higher. Of course, if the pump is single-acting, the deliveries will be one-half those given in above Table.

The following is a useful rule for determining the size of feed pipes of triple and quadruple engines:—

**Rule 156.** $$\text{Diameter feed pipe} = \frac{\sqrt{\text{I. H. P.}}}{K} + 0{\cdot}5 \text{ inch.}$$

For engines having their own feed pumps K = 18; when fed by independently worked pumps K may be 22.

**Feed-Pump Plungers.**—These should be of good bronze, and when of the single-acting ram type may be of thickness given by,—

**Rule 157.**

$$\text{Thickness of metal of plunger} = \sqrt{\frac{P \times D}{4660}} + \cdot 15 \text{ inch,}$$

where P is working pressure in lbs. per square inch, and D is diameter of ram in inches.

**Feed-Tanks.**—To avoid waste of water through the occasional overflow of the hot-well, it is the rule in the Navy to fit a feed-tank, or supplementary hot-well, into which the air pumps deliver, and from which the feed pumps draw; and, where fresh water only is used, such an arrangement is quite necessary.

*The capacity of the tank* is usually about 1 cubic foot for every 20 I.H.P. The tanks are commonly of $^5/_{16}$-inch steel plates, flanged over and single riveted, stayed for a pressure of 10 to 15 lbs. per square inch, and galvanised after completion. Each tank should have a manhole, an air-pipe, an overflow pipe, three or four zinc blocks to prevent corrosion, and a water-gauge giving a visible range of at least 3 feet; and where there are two or more engine-rooms, with a tank in each, the various tanks should be connected one with another. These tanks are frequently built into the structure of the ship. The feed-tanks should be placed as close to the air-pumps and as low down as possible; where they are under the platforms a float and index is more convenient than a water-gauge.

**Reserve Tanks, &c.**—Where the boilers are fed with fresh water only, a sufficient reserve supply should be carried to provide about 21 cubic feet, or say 1300 lbs. "make-up," or auxiliary feed, per 100 I.H.P. per 24 hours.* This supply may either be carried in reserve tanks, in the double bottom or elsewhere; or may be produced as required by evaporators delivering into the L.P. valve-casing, or into the condenser; or, a combination of these two methods may be employed. In recent Naval practice the tanks have a capacity of about 1 c. ft. for every 6 I.H.P.

**Feed Heaters.**—Various types of feed heater are now in common use; and, where space and weight allow of their adoption, they are, without doubt, a very desirable adjunct, since they keep a great deal of deposit out of the boilers, and,—when the exhaust steam from the auxiliary engines can be used in them,—save a considerable amount of heat that would otherwise be wasted.

## Board of Trade Rules relating to Feed Pumps, Pipes, &c.

**Paragraph 136.**—Each boiler of a passenger vessel, whether old or new, should be fitted with suitable check valves between it and the feed pipes, and the boilers of all *new* passenger vessels, should be fitted with separate feeding arrangements in addition to, but unconnected with, the main feed pipes and valves. It is desirable that the main feed check valve chest on each boiler should be separate and distinct from that of the auxiliary feed, and that a stop cock or stop valve should be fitted in each chest or between each chest and the boiler, so that the latter may be shut off, and either of the check valves examined while the other feed is at work. In very small vessels an efficient hand-pump, instead of the usual donkey pump, may be passed if the Surveyor has satisfied himself as to its efficiency when steam is up, and provided that there are separate feed pipes and valves as directed above.

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* With care the *average* consumption may be kept down to about one-half this quantity.

The Surveyor should discourage the practice of using the same pump for the bilges and feeding boilers.

### Lloyd's Rules relating to Feed Pumps, Pipes, &c.

*Sec.* 7. The engines are to be fitted with two feed pumps, each capable of supplying the boilers; the pumps, &c., to be so arranged that either can be overhauled whilst the other is at work.

3. The main feed pumps may be worked by independent engines, provided they are fitted with automatic regulators for controlling their speed. If only one such pump is fitted for the main feed, the auxiliary feed pump required by paragraph 25 should also be fitted with an automatic speed regulator.

6. A steam pump is to be provided capable of supplying the boilers with water, this pump to be provided with suctions from hot-well and from sea. A steam pump is to be so fitted as to pump from each compartment, to deliver water on deck, and, if no hand pump is fitted in engine-room, it must be fitted to be worked by hand. In small vessels in which only one steam pump is fitted, it must comply with all these requirements.

## BILGE PUMPS.

There is no definite basis of calculation for the size of these pumps, and no generally recognised rule. For jet-condensing engines they were generally of the same size as the feed pumps, but for surface-condensing engines the following rule may be used :—

**Rule 158. Capacity of Bilge Pump** $= \dfrac{\text{Capacity of L.P. Cylinder}}{350}$.

The following rule is, however, more appropriate to the requirements, D being the displacement of the ship in tons; there may be from one to four pumps.

**Rule 158a.—Total Bilge Pumps capacity in cubic inches should be not less than** $3 \cdot 5 \times D^{2/3}$.

Where the pumps are driven by the main engines there should be,—except in the case of very small ships,—two pumps of the size given by these rules, arranged so that either may be worked independently of the other, and readily put out of action when not required.

Covers of valve-boxes and mud-boxes should be secured by hinged bolts and wing nuts, in order that they may be easily and quickly removed, and replaced whenever necessary.

All strainer, or mud, boxes should be placed in easily accessible positions *above* the floor plates, as terminal rose-boxes are always troublesome, and may be a cause of serious danger.

The "directing" or "distribution" valve-boxes should also be above the floor plates, and easily accessible; and each cover or hand-wheel should bear a name plate indicating the compartment with which it opens communication; a very good, and also very cheap, plan is to cast the required letters or words on the upper face of the hand-wheel rim.

Where one bilge pump must also act as wash-deck or fire pump (to comply with Board of Trade regulations), a three-way open-bottom cock, with one port in plug, should be fitted, to prevent the possibility of sea-water entering the ship; when the pipes are over three inches in diameter, however, it is not always convenient to fit a cock, and non-return valves may be fitted,—though they are less safe than the cock.

## Board of Trade Regulation referring to Bilge Pipes, &c.

82. **Sounding Pipes.**—Sounding pipes should be fitted from the upper deck for ascertaining the depth of water in each compartment.

The sounding pipes for the ballast tanks under the engine and boiler room and under the tunnel floor may, however, be short, provided that they are fitted with screwed caps at the upper ends, or with cocks having the handles secured to the plugs.

83. **Engine-room Pump Fittings.**—Suction pipes connected with pumps worked by the main and donkey engines should be carried through the bulkheads into all the compartments fore and aft of the engine-room except the compartment in front of the collision bulkhead, so that each compartment can be pumped out separately by the engines. When a double bottom is fitted extending the full length of a compartment, with waterways on each side, a bilge suction should be fitted to each waterway. This requirement need not, however, be enforced in small compartments near the ends of the vessel. The pipes should be well secured where they pass through the bulkheads. In all new steamships, the cocks or valves which are fitted for the purpose of shutting off or controlling the flow of water through these pipes should, unless situated in the compartment occupied by the pump to which the pipes are connected, be provided with means by which they may be manipulated from a height well above the deep load line, preferably the upper deck.

The arrangement of bilge suction pipes should be such that water cannot pass through them from one compartment to another. Valves in bilge distribution boxes should be of the non-return type, and if cocks are used instead of valves, suitable non-return valves should be fitted in the pipes to prevent water passing from one compartment to another in the event of the cocks being left open.

The free end of each hold suction pipe should be fitted with a suitable rose-box or strum, and the tunnel-well suction pipe should be similarly fitted, unless it is provided with a mud-box of the form required for the pipes in connection with the engine-room and stokehold bilges.

Each main and donkey-engine suction pipe in connection with the engine-room and stokehold bilges should be provided with an efficient mud-box, or other similar appliance, placed above the platform, or in any other position in which it will be accessible at all times. The pipe leading from each mud-box to the bilge should, when practicable, be straight, and the cover of the mud-box should be secured in such a manner as to permit of it being expeditiously opened or closed.

The connection from the donkey engine to the bilge main should be by means of a switch cock or non-return valve.

An efficient bilge injection should be fitted to each main circulating pump.

## Lloyd's Rules relating to Bilge Pumps, Pipes, &c.

2. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled whilst the other is at work.

4. A bilge injection, or bilge suction to the circulating pump, is is be fitted.

5. The engine bilge pumps are to be fitted capable of pumping from each compartment of the vessel, the peaks excepted. All bilge suction pipes are to be fitted with strum-boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should be not less than double that of the cross section of suction pipe. The mud-boxes and roses in engine-room are to be placed where they are easily accessible, and to the satisfaction of the Surveyor.

6. A steam pump is to be provided capable of supplying the boilers with water, and is to have suctions from hot-well and sea. A steam pump is to be so fitted as to pump from each compartment, to deliver water on deck, and, if no hand pump is fitted in engine-room, it must be fitted to be worked by hand. In small vessels in which only one steam pump is fitted, it must comply with all these requirements.

10. Bilge suction pipes to be arranged to pump direct from each compartment,—the roses to be fixed in easily accessible places; for numbers and sizes of suctions required, *see* Lloyd's Rules relating to holds of ships, p. 308.

*Sec.* 2.—5. Cocks and valves connecting all suction pipes to be fixed above the stokehole and engine-room platforms.

6. The arrangement of pumps, bilge injections, suction and delivery

pipes is to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect. Any defective arrangement to be reported to the Committee.

See also **Lloyd's Requirements** for sea connections, page 265.

## PUMP LEVERS AND LINKS, &c.

In order to obtain quantitative results for guidance in proportioning pump gear, it is necessary to assume some maximum load per square inch of bucket, and the load here assumed is 30 lbs. for both air and circulating pumps.

**Pump levers.**—The strength of these may be determined by means of the ordinary rule for beams,—

**Rule 159.** $$W=\frac{fz}{l},$$

and,—when the central hole cut in the plate to admit the gudgeon boss does not exceed ·4 × depth of beam, and the attachment of the central boss or gudgeon flange is made with rivets in the usual way,—the values of $f$ may be as follows :—

For thin plates, $f=4500$ for screw engines, and 5500 for paddle engines.

For solid levers (forged) $f=5500$ for screw engines, and 6500 for paddle engines.

In determining $z\left(=\frac{bd^2}{6}\right)$, $b$ is four times the thickness of the plate, for ordinary plate beams, and $d$ is the total depth or width of the plate at the gudgeon.

A lower value of $f$ is adopted for screw engines because, by their higher speeds, there is the liability to severe strains from sudden reversals and loads of the air-pump. When air and circulating pump are both driven by the same pair of levers, the values of $f$ should be reduced 15 per cent,—as one lever of the pair may have to carry considerably more than half the load at times.

**Pump links.**—For the various reasons stated above, the bolts in pump links require to be made very heavy.

**Rule 160.**—The link bolts of paddle engines, when two in number, may carry loads equal to those given in Table XLIII. ; and when four in number, 10 per cent. less, or 20 per cent. less according as air-pump only, or both air and circulating pump, are driven by one pair of levers. The link bolts of screw engines (usually four in number) may, in similar circumstances, carry loads 30 per cent., and 40 per cent below those given in Table XLIII.

The loads that link bolts may carry under the various conditions mentioned will then be as follows:—

## Table LXIV.—Strengths of Pump Links.

| Diameter of link bolt in inches. | Total load in lbs. on one bolt. | | | | |
|---|---|---|---|---|---|
| | Paddle Engine. | | | Screw Engine. | |
| | Two bolts. | Four bolts (air-pump). | Four bolts (air and circulating pumps). | Four bolts (air-pump). | Four bolts (air and circulating pumps). |
| 1 | 2,150 | 1,930 | 1,720 | 1,500 | 1,300 |
| 1⅛ | 3,000 | 2,700 | 2,400 | 2,100 | 1,800 |
| 1¼ | 4,200 | 3,800 | 3,400 | 2,900 | 2,500 |
| 1⅜ | 5,400 | 4,800 | 4,300 | 3,800 | 3,200 |
| 1½ | 7,100 | 6,400 | 5,700 | 5,000 | 4,200 |
| 1⅝ | 8,500 | 7,600 | 6,800 | 6,000 | 5,100 |
| 1¾ | 11,000 | 10,900 | 9,300 | 7,700 | 6,600 |
| 1⅞ | 13,100 | 11,800 | 10,500 | 9,100 | 7,800 |
| 2 | 16,100 | 14,500 | 12,800 | 11,200 | 9,600 |
| 2¼ | 20,400 | 18,300 | 16,200 | 14,200 | 12,200 |

When the links are composed of pairs of parallel bolts, on to the ends of which the brasses are threaded, it will generally suffice to make the diameters of the middle portions of the bolts, between the brasses, the same as those of the end or bolt portions,—since the compressive stresses are usually much less than the tensile.

When the length of the plain or middle portion of the bolt reaches 25 diameters, however, it is advisable to substitute one central rod with T ends for the two parallel ones.

If, in any case, the links are so placed that the compressive stresses exceed the tensile, and if the length of link be more than 10 diameters of bolt or pillar, their size had better be determined by the rules for struts fixed at both ends.

As the stroke of the pumps is commonly less than the stroke of the piston, the load on the links at the piston-rod end of the levers is less than the load on those at the pump end in inverse proportion to their distances from the fulcrum of the lever.

**Surfaces of pins.—Rule 160a.**—The surfaces of the various pins (diameter × length) to which the pump links are attached, should be such that the load does not exceed $\frac{18,000}{\sqrt{Rd}}$ lbs. per square inch, R being the revolutions and $d$ the diameter of pin ; generally the working load should be 70 per cent. of this maximum.

**Pump crossheads.**—These should be calculated as beams loaded in accordance with the circumstances of the case.

Considering the sections usually employed, and the nature of the load, the value of $f$ for mild steel may be taken at 8600. For further guidance see general formulæ for beams.

**Pump lever gudgeons.**—The load on the pump lever gudgeon, and on its bearings, cap bolts, &c., is the sum of the loads on the two ends. The cap bolts should not carry greater loads than link bolts of corresponding diameters. (*See* Table LXIV., page 176.)

The bearings may carry a load not exceeding $\frac{16,000}{\sqrt{Rd}}$ lbs. per square inch. The diameter of gudgeon will be most conveniently determined by considering the stresses to which it is subject as shearing stresses simply; then,—

**Rule 161.** **Diameter of Gudgeon** $= \sqrt{\frac{S}{K}}$

where S = the sum of the total loads on the two ends of the pair of levers;

K = 1200,—when there are two bearings, one close to each lever, as is usual in screw engines;

K = 1600,—when there is only one lever in place of a pair, with very short gudgeon and bearing at each side close to lever, as frequently fitted in paddle engines.

## SLIDE VALVES.

**Travel of Valve.—Rule 162.**—The travel of a slide-valve should be proportional to the length of the steam-port in cylinder face, measured parallel to axis of slide-rod, and not less than twice the amount.

As the work done in moving the valve is nearly proportional to the length of travel, it is important to keep this latter as short as possible, and the best way of effecting this is to use double- or triple-ported valves.

Single-ported valves should not be used for cylinders of more than 25 inches diameter; whilst triple-ported valves can only be conveniently employed for large engines of fairly long stroke, and are therefore only applicable in Naval work when the piston speed is very high and the cylinders large.

Since the introduction of the triple engine, piston valves have been reintroduced, and are always used for H.P. cylinders, and generally for the M.P. also; their worst feature is their liability to stick when any dirt or grit is deposited on them, and the consequent severe strains that may be thrown on the valve-gear; also, as ordinarily fitted, they are not so steam-tight as flat valves.

**Surface of Valve.**—Flat slide-valves sometimes give unsatisfactory results from want of sufficient bearing surface,—so that the pressure per square inch of rubbing surface is so great that undue wear, cutting, &c., result.

**Relief-rings.**—Flat slide-valves should be fitted with some arrangement for relieving the pressure on the back, to reduce wear and tear, and save the driving power; they also require a spring of some sort to replace them against the face, after they have been forced from it by the presence of water in the cylinder, or by other causes.

**Rule 163.**—The pressure per square inch of rubbing surface, imposed by the springs, should not exceed 2½ lbs.

Small valves have one relief-ring, and very wide ones two rings side by side; the areas may be made as large as can be got on the valve backs, when the relief pipe from M.P. valve is led to L.P. receiver, and that from L.P. valve to condenser in the usual manner; a greater difference of pressure than given by this arrangement is not desirable, and any leakage will be a serious loss.

**Port openings.**—With given leads and cuts-off, both reckoned as percentages of the stroke of piston, the openings to steam are proportional to the travels, and may therefore be increased or diminished by increasing or diminishing the travel.

**Lead of valve.**—The lead given to the valve is generally decided arbitrarily, and varies from a bare 1/32 inch in small auxiliary engines to 1¼ inch in large L.P. valves; but the piston speed and momentum of moving parts should be taken into consideration, as in M.P. and L.P. cylinders the compression is rarely sufficient in itself to absorb all the momentum, and a certain amount of lead is necessary to prevent shock.

The extent to which the engine will probably be run "notched" or "linked" up should also be considered.

Ordinarily, when the valves are properly set, the lead at the back or top end will be about half that at the front or bottom end, and the cut-off at the crank end will be earlier than at the other, but the cut-off may be the same at each end if the leads are different. With a valve cutting off late, inside lap is necessary to give the compression sufficient for quiet working; but it is a means of throttling, as well as delaying the period of exhaust. With an early cut-off sufficient compression can be obtained and a freer exhaust enjoyed if there is negative lap.

For a quick-running compound or triple engine, exhaust from H.P. cylinder should commence at ·85 of the stroke, and for a slow-working paddle engine, not later than ·95 of the stroke.

**Proportions of slide-valves.**—Figures 26 and 27 show some of the elementary proportions of the common and of the Trick valve.

Let $x$ be the *outside* lap of the valve at the front end, and $y$ that at the back end. Then (Fig. 26),—

$$H=\frac{G}{2}+x\text{; and }K=\frac{G}{2}+y.$$

Also, let $z$ be the *inside* lap at the front, and $w$ that at the back. Then (Fig. 26),—

$$B=\frac{F}{2}-z\text{; and }C=\frac{F}{2}-w.$$

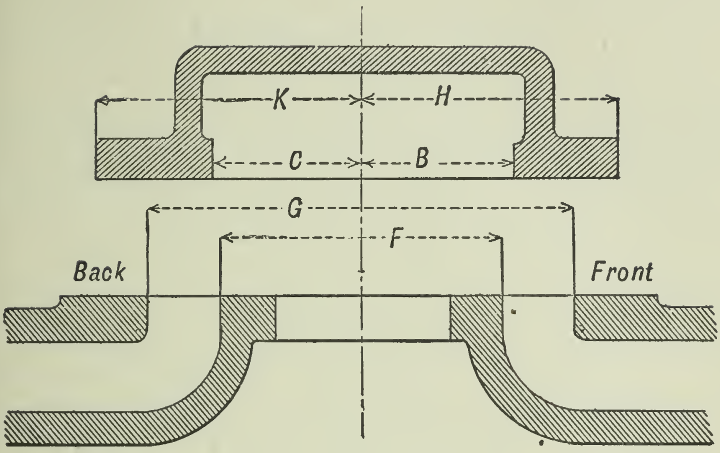


FIG. 26.—Proportions of a Common Valve.

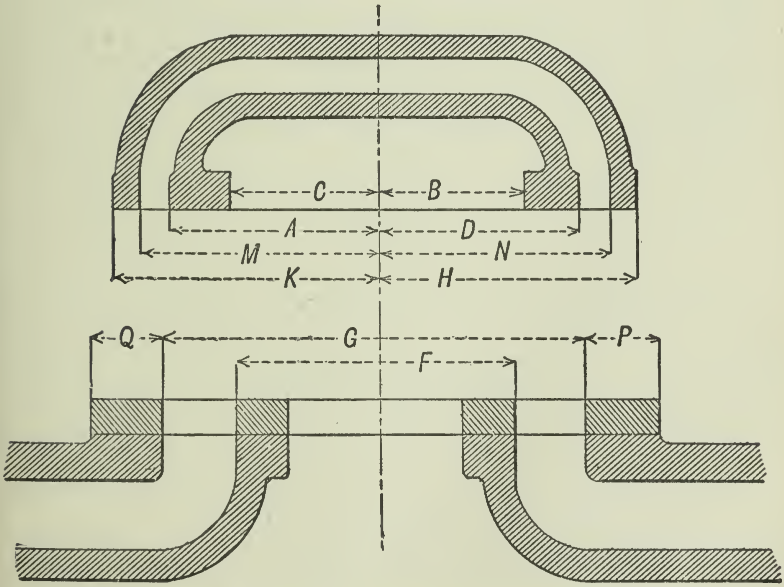


FIG. 27.—Proportions of a Trick Valve.

Referring now to Fig. 27,—let $x$, $y$, $z$, and $w$ be the laps as before. Then,—

$$H=\frac{G}{2}+x\text{; and }K=\frac{G}{2}+y.$$

Also,—

$$B=\frac{F}{2}-z\text{; and }C=\frac{F}{2}-w.$$

$$A=\frac{G}{2}+\tfrac{1}{8}\text{ inch; and }D=\frac{G}{2}+\tfrac{1}{8}\text{ inch.}$$

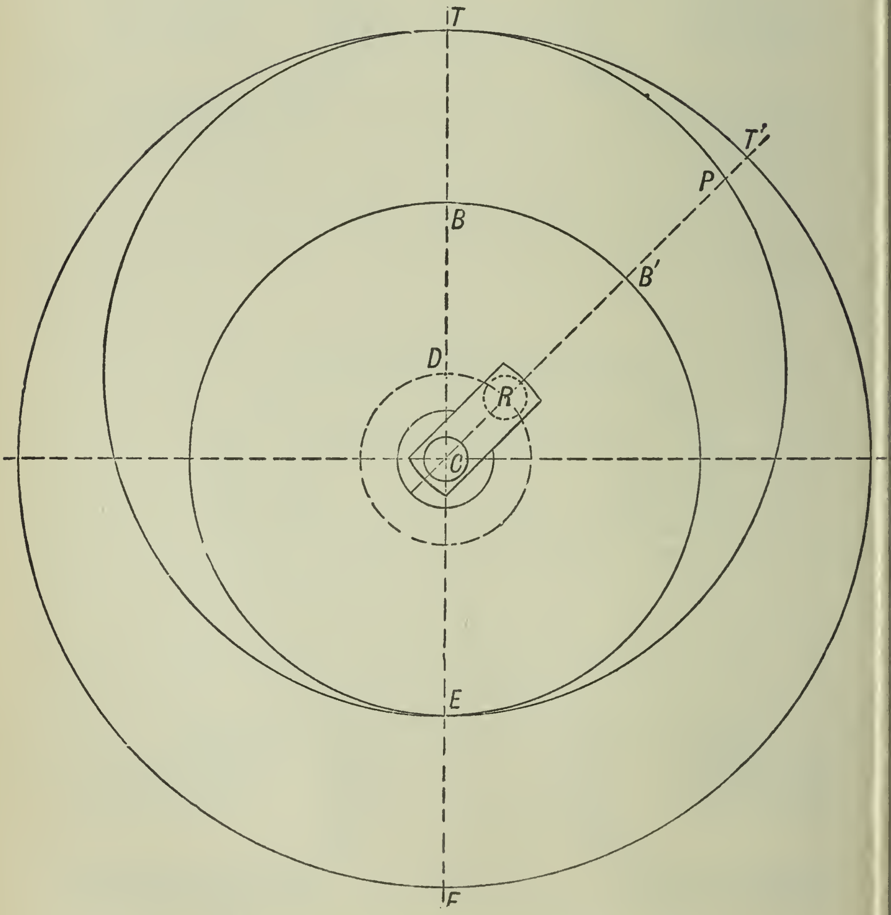


Fig. 28.—Diagram of the Piston Path.

The openings through the valve laps or covers must be as large as possible, but need not exceed the ordinary opening of the valve to steam at the outer edge; then,—

$$G+P=K+N; \text{ and } G+Q=H+M.$$

**Valve diagrams.**—Figures 28 and 29, when combined and used as described below, show at a glance the complete cycle of the operation of any proposed valve, and also the effect of varying any one of the elements,—travel, laps, leads, or openings.

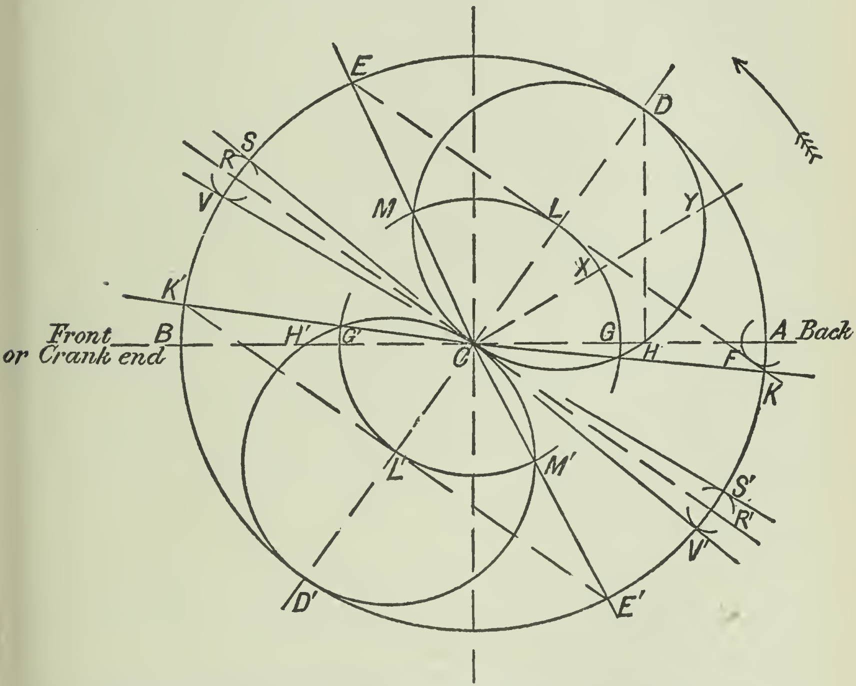


Fig. 29.—Zeuner's Diagram for the Common Valve Motion.

In constructing the diagram for a proposed valve, the diagram Fig. 29, the extreme diameter of which is equal to the travel of the valve, should be placed in the centre of the diagram Fig. 28,—the former being drawn full size, and the latter ¾ inch or 1 inch to the foot, as found most convenient.

Referring to Fig. 28, the outer circle TT′F represents the top or back end of the cylinder; the inner circle BB′E represents the bottom, front, or crank end; and the eccentric circle gives the position of the piston corresponding to any angular position of crank. The diagram is drawn by setting off CD equal to crank, and DT equal to connecting-rod, and then swinging DT round on D as a centre.

T is the position of piston on top or back centre, and E its position on bottom or front centre; and the position corresponding to any other position of crank,—such as CR,—is found by producing CR to cut the circles, when PT′ is the distance of the piston from one end of cylinder, and PB′ its distance from the other.

Referring now to Fig. 29, the problem most frequently met with is,—Given the travel, leads, and cuts-off, to determine the laps and position of eccentric,—and it is solved as follows:—

Draw CE so that, when produced, it will cut the eccentric circle at a point corresponding to the given point of cut-off (*i.e.*, if cut-off is to be at $x$ inches from the back end of the stroke, the point in the eccentric circle must be $x$ inches from the outer circle); then ACE is the angle through which the crank must move to arrive at the position of cut-off. With A as centre, and radius AF equal to the lead, draw part of a circle, and then draw EK. Next, draw CD perpendicular to, and bisecting EK in L. Then CL is the lap required, and BCD is the angle between the crank and eccentric; and since CD is the half-travel, LD is the maximum opening of port.

To extend the usefulness of the diagram,—on CD as diameter, describe a circle; and from centre C with radius CL, strike the arc GLM, which is called the lap circle. The part GH is then equal to AF, and represents the lead, or opening of valve when the crank is "on the centre," or in the position CA. XY likewise represents the port opening when the crank is in position CY.

To determine the operation of the valve at the other end, produce DC to D′, and describe a circle on CD′. Then let H′G′ be the lead at this end; from C, with radius CG′, draw the lap circle G′L′M′, and through C and M′ draw CE′. Then CE′ is the position of the crank at cut-off, and CG′ is the lap.

The positions of the crank, when the port commences to open, are CK and CK′.

If the valve has no inside lap (*i.e.* if the ports are both just closed to exhaust when the valve is in mid-position), release at one end and compression at the other will commence simultaneously when the crank reaches the positions CR and CR′, which are at right angles to DD′.

If, however, the valve has positive inside lap at the front end, and negative inside lap of equal amount at the back end, the positions of release and commencement of compression will be altered as indicated, back release and front compression occurring at S, and front release and back compression at S′; or, if laps be reversed, at V and V′.

The points S, S′ and V, V′ are obtained by striking the small arcs indicated with the inside lap as radius; of course, if the lap is positive, the valve will open for exhaust later and close earlier; but if the lap is negative, the exhaust will be earlier and the compression commence later.

To obtain the position of piston corresponding with the various crank positions, each radial line indicating a crank position must be continued until it meets the outer circle TT′F, and the position of piston can then be scaled off.

**Effect of linking up.**—The effect of "linking up" on the operation of the valve is most readily exhibited by the application to the above diagrams of the following very closely approximate construction, suggested by the late Mr Macfarlane Gray :—

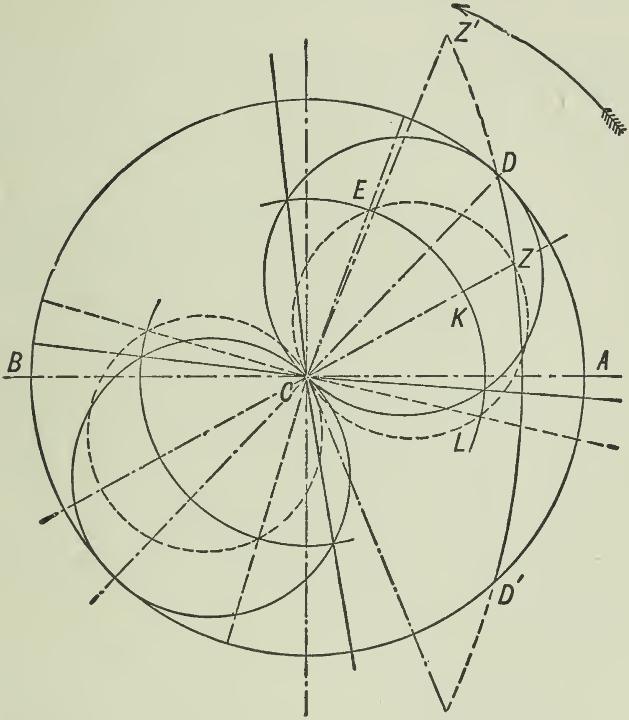


FIG. 30.—Diagram showing the effect of "Notching up."

Suppose the link (Fig. 31) to be notched up so that the link block M is distant MT from the point at which the eccentric rod is attached to the link.

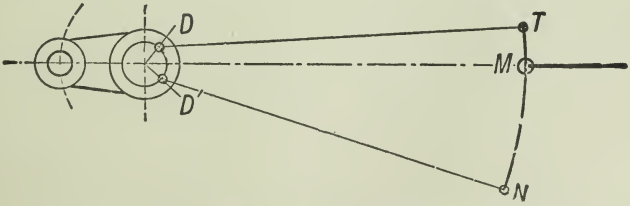


FIG. 31.—Link Motion "Notched up."

Draw the valve diagram (Fig. 30) due to the position and throw of the eccentrics in Fig. 31, and through D and D′ draw the arc of a circle with a radius found as follows :—

**Rule 164.** $$\text{Radius} = \frac{DT \times DD'}{2TN} \text{ (see Fig. 31).}$$

Then divide this arc at Z so that DZ is to DD′ as TM is to TN; join CZ, and on it as diameter describe a circle cutting the lap circle in L and E; and draw the lines CL and CE, the former indicating position of crank when valve opens, and the latter the position when valve closes.

It will thus be seen that the effect on the valve of notching the link up to the point M is the same as though it were driven by an eccentric having the angular position and eccentricity CZ; the lead is earlier, the opening is reduced to KZ, the cut-off is earlier, and a further examination would show that release occurs earlier, and compression commences earlier than when in full gear.

**Open and crossed rods.**—When the eccentric rods are arranged as shown in Fig. 31, they are said to be "open," but when the gear is so arranged that D is joined to N, and D′ to T, whilst the crank remains turned away from the link TN, the rods are said to be "crossed"; in the diagram for this latter case the arc DD′ must be drawn convex towards C, or its centre must be on side A.

When it is intended to work the engines linked up to any considerable extent, the rods should be of the "open" type, as a greater range of expansion can be obtained with less reduction of port openings than is possible with "crossed" rods.

**Overhung gear.**—Where the valve gear is of the "overhung" type (*i.e.* the type in which T is always outside of M, even in full gear), the diagrams are constructed on the same principle as Fig. 30, and the position and eccentricity of the eccentrics are determined by producing the arc DD′, and taking a point Z′ *beyond* D, so that Z′D is to (DD′+2Z′D) as TM is to TN, and joining CZ′. CZ′ is the required eccentricity of the sheaves, and BCZ′ is the angle between them and the crank.

**Obliquity of eccentric rods.**—The valve diagram, as above described, takes no account of the effect produced by the obliquity of the eccentric rods; when the rods are so short as to make this disturbing effect noticeable, the diagram may be corrected as follows :—

From A and B (Fig. 32) drop perpendiculars AN and BN′ upon RR′, and through N and N′, with length of eccentric rod as radius, strike an arc; also, with same radius, strike similar arcs through O and P and O′ and P′. Then, if the points where these arcs cut the travel circle be joined with C, the corrected crank positions, at which the various events of the cycle occur, will be obtained; the radial lines indicating these new crank positions may then be produced outwards so as to cut the eccentric circle and indicate the piston positions as before. The correcting arcs for the exhaust laps have been omitted as tending to complicate the diagram, but they will be drawn on either side of the arc NN′, at distances from it equal to the laps, and will give new points on the travel circle in a precisely similar way.

When the eccentric rods are very short, and the locomotive or "slot" link is used, as in some types of horizontal engine, no diagram will give more than approximately accurate results, and the final adjustments should be made on a model: one that will serve every purpose may be rigged up on a drawing-board with a few pieces of lath and some stout pins, in little more than half an hour.

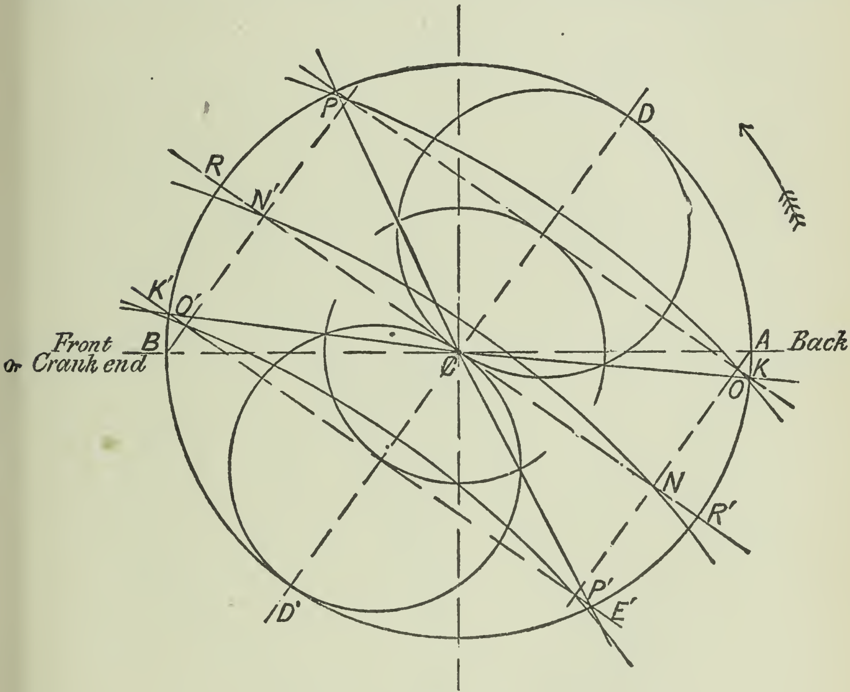


FIG. 32.

For the solutions of other problems connected with the diagram for the ordinary slide-valve, and for the methods of constructing diagrams for various types of expansion valve, see Mr Seaton's "Manual of Marine Engineering."

**Diagram for oscillating engine.**—When the diagram Fig. 28 is applied to the case of an oscillating engine, the eccentric circle deviates slightly from a true circle, owing to the fact that there is virtually a connecting-rod of varying length. The necessary correction in the eccentric circle may be made as follows :—

Let A be the trunnion centre, and BLE the crank path ; divide the half-crank path into any number of equal parts (say six), at K, H, G, &c. ; join each point with A ; and from A, with radius AB, strike the arc BDC. Then, referring to Fig. 34, set off the corresponding points E, F, G, &c., and transfer the lengths ED, FD, GD, &c., from Fig. 33 to the positions indicated by the corresponding letters on Fig. 34, and through the points D, D, D, &c., draw a curve ; this curve will then be the true curve of successive positions of piston corresponding to the crank positions indicated by any radial lines.

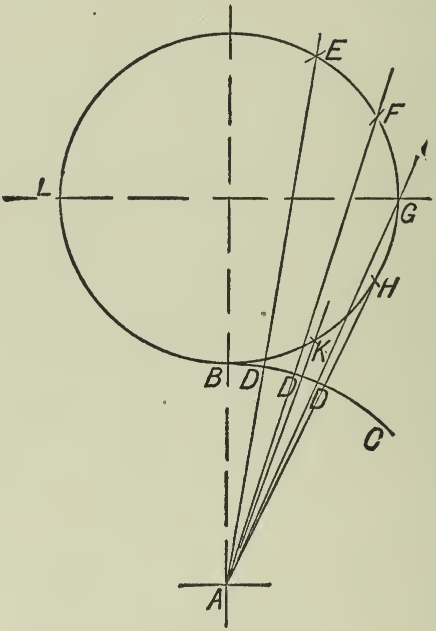


Fig. 33.

GL in Fig. 34 (p. 187) must, of course, represent GL in Fig. 33, either to the same or some other convenient scale.

## VALVE GEAR.

It must be understood that the various proportions given in the following sections on "Valve Gear" and "Eccentrics" are such as are suitable for *ordinary* speeds of engines of the two categories, Naval and Mercantile, and that both increased strengths and increased surfaces will be required if the engines are to run at exceptionally high numbers of revolutions.

**Valve or slide rods.**—The power requisite to move a slide-valve on its face is a constantly varying quantity, and care must be taken to use its maximum value in all calculations for dimensions of valve gear.

To obtain this, the reduction due to any relief arrangement, or to the pressures on some portions being in equilibrium, should be neglected, and the total pressure on the valve taken as the measure of load, acting on its gross area,—these pressures being those of the steam-chest or receiver on one side, and those of receiver or condenser on the other.

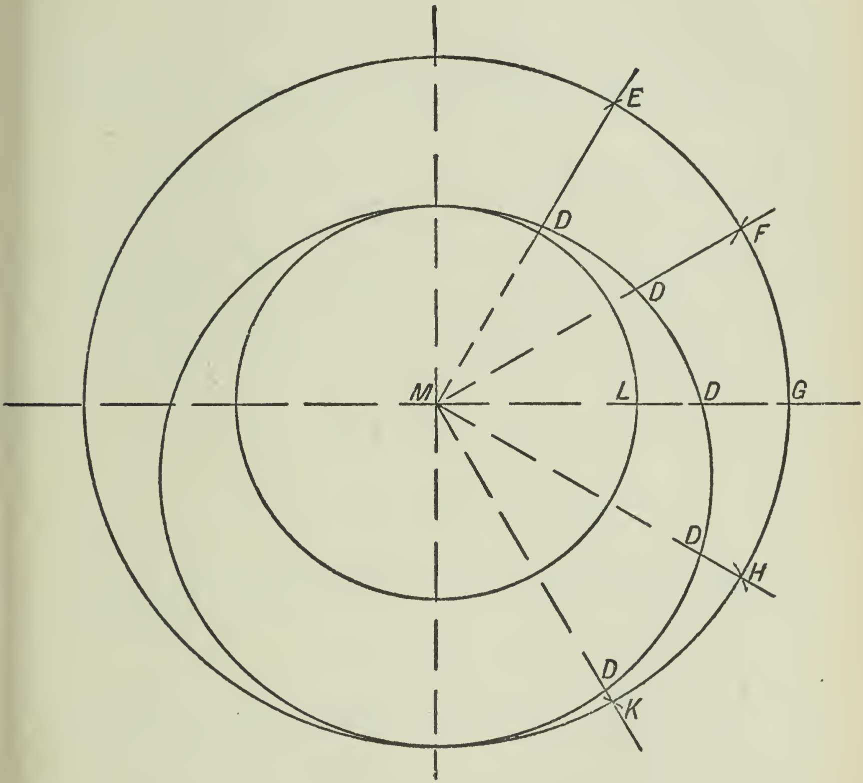


FIG. 34.

If L be the length and B the breadth of a valve, both in inches; $p$ the maximum pressure per square inch, obtained as described above, and ·2 the co-efficient of friction for metallic surfaces rubbing together dry; then,—

**Rule 165.** Load on valve-rod $= \cdot 2\,(L \times B \times p)$ lbs.;

and,—**Rule 166,**—

Smallest diameter of valve-rod $= \sqrt{\dfrac{L \times B \times p}{12500}} + \cdot 25$;

and,—**Rule 167,**—

Diameter of valve-rod in gland $= \sqrt{\dfrac{L \times B \times p}{6500}} + \cdot 25$.

The value of $p$ should be taken as 30 lbs. for L.P. valves, and, as it is desirable to have the rods all of the same dimensions, the highest value of the above expression for any one cylinder should be used for all.

**When piston-valves** are used for the H.P. cylinders, the valve-rod is usually made of the same size as those for the flat slides of the M.P. and L.P. cylinders.

Theoretically, the power required to move a piston-valve should be little more than is necessary to overcome the inertia and friction at the stuffing-box, but practically,—as mentioned under "Slide-valves,"—the power required is in excess of this, and may be largely so.

The diameter of rods for piston-valves may be determined by the following:—

**Rule 168.** Diameter rod $= \dfrac{\text{Diameter valve} \times \sqrt{p}}{F} + \cdot 6$ in.

$p$ for H.P. cylinders is the boiler-pressure.
$p$ ,, M.P. ,, ,, half ,,
$p$ ,, L.P. ,, ,, 30 lbs. ,,

For very light engines, as in destroyers, $F = 100$.
For express steamers, short service, and Naval ships generally, $F = 90$.
For merchant ships generally, $F = 84$.

In practice the rod of the H.P. valve is about $\frac{1}{6}$th the diameter of the valve; that of the M.P. about $\frac{1}{8}$th; and that of L.P. cylinder $\frac{1}{10}$th.

For convenience the valve-rods are of one size throughout.

**Rule 169. The cap-bolts** for valve-rod end or eye must each carry a load equal to $\dfrac{\cdot 2\,(L \times B \times p)}{2}$, and the size suitable for this load will be found from Table XLIII., page 103. Their diameter will be found to very closely approach $\cdot 5D + \cdot 2$ inch, where D is diameter of valve-rod found by Rule 166 above.

**Valve-rod guides.**—All valve-rods should be fitted with guides, placed as close to the link as possible, and, if not fitted with balance pistons, they should have guides at the tail ends as well.

The most satisfactory form of guide is one resembling a piston-rod slipper guide of the "single" type (*i.e.* one shoe and one guide for both ahead and astern) in miniature,—the shoe being cast with a cylindrical crosshead which is bored to fit the valve-rod, and secured on it by a through cotter.

When this type is not adopted, and the guide is made to embrace the valve-rod just below, or outside of, the gland, a square or two flats should (whenever space is available) be formed on the rod to take the guide.

**Reversing quadrants, or "links."**—The old slot-link still retains its place, as the most convenient of application in oscillating paddle engines, and in some types of screw engines. For small fast-running engines it is quite the safest link and still retained by locomotive engineers.

Its principal proportions should be as follows—the unit D being the diameter of valve-rod as found by Rule 166.

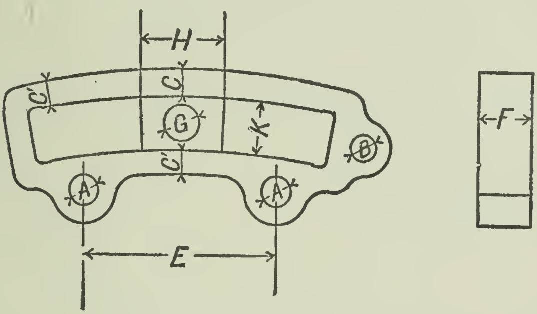


FIG 35.

**Rule 170, a to k.**

| | | | |
|---|---|---|---|
| (*a*) Centres of eccentric-rod pins (E) | | 2·75 to 3 × throw of eccentrics. | |
| (*b*) Breadth of link (F) | | 1·1 D + ·4 | inch. |
| (*c*) Thickness of bar (C) | with bridge | D + ·25 | ,, |
| | without ,, | 1·1 D + ·25 | ,, |
| (*d*) ,, ,, (C′) | with ,, | ·9 D + ·25 | ,, |
| | without ,, | D + ·25 | ,, |
| (*e*) Length of block (H) | | 2·5 D + 1 | ,, |
| (*f*) Width of slot (K) | | 1·4 G + ·35 | ,, |
| (*g*) Diam. of eccentric-rod pin (A) | solid bush | D + ·25 | ,, |
| | adjustable brasses | 1·1 D + ·25 | ,, |
| (*h*) , drag-rod pin (B) | as in fig. | ·55 D + ·25 | ,, |
| | when on bridge | ·8 D + ·25 | ,, |
| (*j*) ,, block-pin (G) | when overhung | 1·4 D + ·25 | ,, |
| | both ends secured | 1·3 D + ·25 | ,, |
| (*k*) ,, drag-rod at ends | if one | ·7 D + ·25 | ,, |
| | if two | ·48 D + ·25 | ,, |

Theoretically, surfaces may be a little smaller for paddle engines, but it is better to give the extra surface and make them same as for screw engines.

For all vertical engines, and in all other cases where possible, excepting where L.P. cylinder is below, say, 30 inches in diameter, the double-bar link is preferable, as large working surfaces and pins are readily obtained, and they are more accessible and easier to adjust.

The principal proportions of the double-bar link should be as follows, —the unit being again D, or diameter of valve-rod as found by Rule 166.

**Rule 171, a to n.**

| | |
|---|---|
| (*a*) Centres of eccentric-rod pins | 3 × throw of eccentrics, or roughly 7·8 D. |
| (*b*) Depth of bars | 1·4 D + ·75 inch. |
| (*c*) Thickness of bars | ·5 D + ·15 ,, |
| (*d*) Width between bars and thickness of valve-rod eye when gun-metal or bronze block | 1·2 D + ·25 ,, |
| (*e*) Width between bars when cast-steel block and gun-metal liners | 1·65 D + ·25 ,, |
| (*f*) Dia. of valve-rod eye (G.M. or bronze block) | 1·55 × depth of link-bar. |
| (*g*) ,, ,, (steel block and G.M. liners) | 1 × ,, ,, |
| (*h*) Length of sliding block | 3 D to 3·75 D. |
| (*j*) Diameter of eccentric-rod pins | ·85 D + ·5 inch. |
| (*k*) Length ,, ,, | ·7 D + ·4 ,, |
| (*l*) Diameter of drag-rod pins | ·7 × diameter of eccentric-rod pin. |
| (*m*) Length ,, ,, | 1 × length of eccentric-rod pin. |
| (*n*) Diameter of bolt through ends of link bars | ·35 × depth of link bar. |

For proportions given by above rules, *see* Table LXV..

The links and all the pins and working surfaces should be thoroughly case-hardened before completion if not made of hard steel.

**Position of suspension pins.**—It is convenient to place the suspension, or drag-rod pins at the end, in the case of the slot link, but, when the engine has to work equally ahead or astern, they are better placed at mid-length of the link,—so as to coincide in position with the block-pin when the link is in mid-gear,—so as to give the least "slotting" motion.

The reversing lever when in mid-position was commonly parallel to the valve-rod, but it is better to incline it slightly away from the valve-rod, and so reduce the "slotting" when in ahead gear, whilst increasing it a little when in astern gear. The amount of inclination, measured on the arc, may be about one-eighth the length of the lever.

The drag-rods should, of course, be as long as they can conveniently be made, and care should be taken that the angle they make with the reversing lever is not too obtuse when at its maximum.

The position of the pin of reversing lever for the least amount of slotting can be found by placing the link in "ahead" gear and

## Table LXV.—Proportions of Double-Bar Links.

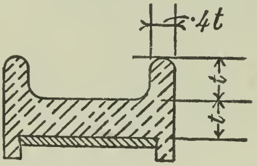
| Diameter of valve-rod (in valve). | Centres of eccentric rod pins, (about) (*a*) | Depth of bars. (*b*) | Thickness of bars. (*c*) | Width between bars, G.M. block. (*d*) | Width between bars, steel block. (*e*) | Diameter of valve-rod eye, G.M. block. (*f*) | Diameter of valve-rod eye, steel block. (*g*) | Length of block on bars, (3·5 D) (*h*) | Diameter of eccentric rod pins. (*j*) | Length of eccentric rod pins. (*k*) | Diameter of drag-rod pins. (*l*) | Length of drag-rod pins. (*m*) | Diameter of end bolt. (*n*) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. |
| 1½ | 12 | 2⅞ | ⅞ | 2 | ... | 4½ | ... | 5¼ | 1¾ | 1½ | 1 | 1½ | 1 |
| 1¾ | 14 | 3¼ | 1 | 2⅜ | ... | 5 | ... | 6 | 2 | 1⅝ | 1⅛ | 1⅝ | 1⅛ |
| 2 | 16 | 3½ | 1⅛ | 2⅝ | ... | 5½ | ... | 7 | 2¼ | 1¾ | 1¼ | 1¾ | 1¼ |
| 2¼ | 18 | 4 | 1¼ | 3 | ... | 6¼ | ... | 7¾ | 2⅜ | 2 | 1⅜ | 2 | 1⅜ |
| 2½ | 20 | 4¼ | 1⅜ | 3¼ | ... | 6¾ | ... | 8¾ | 2⅝ | 2⅛ | 1½ | 2⅛ | 1½ |
| 2¾ | 21 | 4⅝ | 1½ | 3½ | 4¾ | 7¼ | 4⅝ | 9½ | 2¾ | 2⅜ | 1⅝ | 2⅜ | 1⅝ |
| 3 | 23 | 5 | 1⅝ | 3⅞ | 5¼ | 7¾ | 5 | 10½ | 3 | 2½ | 1¾ | 2½ | 1¾ |
| 3¼ | 25 | 5¼ | 1¾ | ... | 5⅝ | ... | 5¼ | 11¼ | 3¼ | 2⅝ | 1⅞ | 2⅝ | 1⅞ |
| 3½ | 27 | 5⅝ | 1⅞ | ... | 6 | ... | 5⅝ | 12¼ | 3½ | 2⅞ | 2 | 2⅞ | 2 |
| 3¾ | 29 | 6 | 2 | ... | 6½ | ... | 6 | 13 | 3¾ | 3 | 2⅛ | 3 | 2⅛ |
| 4 | 31 | 6⅜ | 2⅛ | ... | 6⅞ | ... | 6⅜ | 14 | 4 | 3¼ | 2¼ | 3¼ | 2¼ |
| 4¼ | 33 | 6¾ | 2¼ | ... | 7¼ | ... | 6¾ | 14¾ | 4⅛ | 3⅜ | 2⅜ | 3⅜ | 2⅜ |
| 4½ | 35 | 7 | 2⅜ | ... | 7⅝ | ... | 7 | 15¾ | 4¼ | 3½ | 2½ | 3½ | 2⅜ |
| 4¾ | 37 | 7⅜ | 2½ | ... | 8 | ... | 7⅜ | 16½ | 4½ | 3¾ | 2⅝ | 3¾ | 2½ |

*Note.*—In Naval work, owing to the higher speeds of revolution used, the proportions and surfaces are usually somewhat greater than in Mercantile work; the above Table gives good average proportions. For paddle engines the surfaces may be same as for screw engines, since little is saved by reducing them.

tracing the *locus* of the centre of the suspension pin when there is no slotting motion whatever. With the length of drag-rod as radius, draw an arc through the figure so traced so as to divide it thereby as uniformly as possible. The centre of this arc is the position of the lever pin. Carry out the same operation for the link in "astern" gear and the other extreme position of the lever end is found. From these two positions the centre of weigh-shaft can be located.

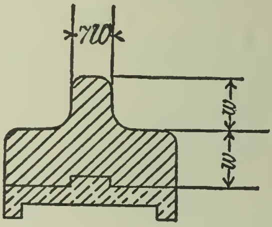
## ECCENTRICS.

**Eccentric Sheaves.**—These should be made of hard and tough cast-iron, and, when made in two pieces, the small piece should be of specially strong iron or steel. The line of division should always be at right angles to the line through centres of shaft and of eccentric, and the key-way should be placed in the centre of the large piece. The joint separating the two portions should also have a small step in it, on each side of the shaft, to prevent their shifting.



*Bronze Strap lined with White-metal.*

FIG. 36.



*Cast-steel Strap with Bronze liner.*

FIG. 37.

The bolts holding the two portions together may either have suitable nuts fitted into recesses in the large portion, or cotters placed where the sheave is perforated.

**Eccentric straps.**—These may be of good bronze for sheaves on shafts up to 10 inches in diameter, but for larger eccentrics they should be of cast-steel, with white-metal or bronze liners. In the mercantile marine good cast-iron, with or without white-metal linings, works quite satisfactorily.

The straps should always have side lips, as shown in Figs. 36 and 37, and when bronze liners are used they should have external projecting rings fitting into grooves in the cast-steel straps, as shown in Fig. 37.

When bronze straps are used, each half should be made thicker at the middle, to allow for wear; and whether bronze or steel is used, the strap should be stiffened by ribs as indicated in Figs. 36, 37, and 38,—one central rib being used for steel, and two side ones for gun-metal.

The general proportions of sheave and strap should be as follows :— the unit D being still the diameter of valve-rod, as found by Rule 166 :—

**Rule 172, a to m.**

| | | | |
|---|---|---|---|
| (*a*) | External diameter of sheave | | 1·2 × diameter of (shaft + throw of eccentric), or roughly 7·8 D. |
| (*b*) | Breadth of sheave at strap | Mercantile | D + ·6 in. |
| | | Naval | D + 1·25 ,, |
| (*c*) | Thickness of metal round shaft | pierced sheave | ·6 D + ·5 ,, |
| | | lightened ,, | ·45D + ·25 ,, |
| (*d*) | ,, ,, at circumference | pierced ,, | ·4D + ·5 ,, |
| | | lightened ,, | ·4D + ·25 ,, |
| (*e*) | Breadth of key | | ·55D + ·4 ,, |
| (*f*) | Thickness ,, | | ·27D + ·3 ,, |
| (*g*) | Diameter of bolt connecting portions of sheave | | ·45D + ·25 ,, |
| (*h*) | Thickness of bronze straps at middle (ex. lips) (*t*) | | ·4D + ·5 ,, |
| (*j*) | ,, ,, ,, sides ,, | | ·33D + ·5 ,, |
| (*k*) | ,, cast-steel straps (*w*) | | ·3D + ·5 ,, |
| (*l*) | ,, bronze liners for cast-steel straps. | $\frac{\text{Dia. of sheave}}{70}$ | + ·125 ,, |
| (*m*) | Diameter of bolts connecting parts of strap | | ·5D + ·2 ,, |

For proportions given by above rules, *see* Table LXVI.

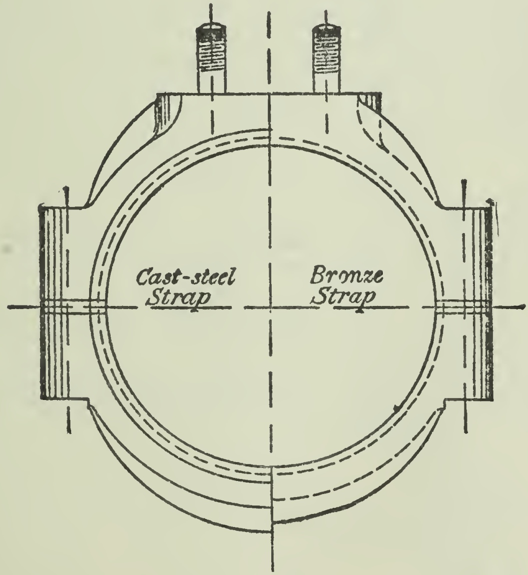


FIG. 38.

Table LXVI.—Proportions of Eccentric Sheaves and Straps.

| Diam. of valve-rod (in valve). | External diam. of sheave (about) | Breadth of sheave at strap. (b) | | Thickness of metal round shaft. (c) | | Thickness of metal at circumference. (d) | | Dimensions of Key. (e) | (f) | Diam. of bolt connecting sheave. | Thickness of bronze straps. (h) | (j) | Thickness of cast-steel straps. | Thickness of liners for steel straps. | Diam. of bolt connecting strap. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Mercantile. | Naval. | Pierced | Lightened. | Pierced | Lightened. | Br'dth | Thickness. | (g) | Middle. | Sides. | (k) | (l) | (m) |
| ins. | ins. | ins. | ins. | ins. | ins. | ins. | ins. | ins. | in. | ins. | ins. | ins. | in. | in. | ins. |
| 1½ | 12 | 2⅛ | 2¾ | 1½ | 1 | 1⅛ | ⅞ | 1¼ | ¾ | 1 | 1⅛ | 1 | 1 | 5/16 | 1 |
| 1¾ | 14 | 2⅜ | 3 | 1⅝ | 1⅛ | 1¼ | 1 | 1⅜ | 13/16 | 1⅛ | 1¼ | 1⅛ | 1 1/16 | 5/16 | 1⅛ |
| 2 | 16 | 2⅝ | 3¼ | 1¾ | 1⅛ | 1⅜ | 1⅛ | 1½ | ⅞ | 1⅛ | 1⅜ | 1 3/16 | 1⅛ | ⅜ | 1¼ |
| 2¼ | 18 | 2⅞ | 3½ | 1⅞ | 1¼ | 1⅜ | 1⅛ | 1⅝ | 15/16 | 1¼ | 1⅜ | 1¼ | 1 3/16 | ⅜ | 1⅜ |
| 2½ | 20 | 3⅛ | 3¾ | 2 | 1⅜ | 1½ | 1¼ | 1¾ | 1 | 1⅜ | 1½ | 1⅜ | 1¼ | 7/16 | 1½ |
| 2¾ | 21 | 3⅜ | 4 | 2⅛ | 1½ | 1⅝ | 1⅜ | 1⅞ | 1 1/16 | 1½ | 1⅝ | 1 7/16 | 1 5/16 | 7/16 | 1⅝ |
| 3 | 23 | 3⅝ | 4¼ | 2⅜ | 1⅝ | 1¾ | 1½ | 2 | 1⅛ | 1⅝ | 1¾ | 1½ | 1 7/16 | 7/16 | 1¾ |
| 3¼ | 25 | 3⅞ | 4½ | 2½ | 1¾ | 1⅞ | 1⅝ | 2¼ | 1 3/16 | 1¾ | 1⅞ | 1⅝ | 1½ | ½ | 1⅞ |
| 3½ | 27 | 4⅛ | 4¾ | 2⅝ | 1⅞ | 2 | 1¾ | 2⅜ | 1¼ | 1⅞ | 2 | 1 11/16 | 1 9/16 | ½ | 2 |
| 3¾ | 29 | 4⅜ | 5 | 2¾ | 1⅞ | 2 | 1¾ | 2½ | 1⅜ | 1⅞ | 2 | 1¾ | 1⅝ | 9/16 | 2⅛ |
| 4 | 31 | 4⅝ | 5¼ | 3 | 2 | 2⅛ | 1⅞ | 2⅝ | 1 7/16 | 2 | 2⅛ | 1⅞ | 1¾ | 9/16 | 2¼ |
| 4¼ | 33 | 4⅞ | 5½ | 3⅛ | 2⅛ | 2¼ | 2 | 2¾ | 1½ | 2⅛ | 2¼ | 1 15/16 | 1 13/16 | ⅝ | 2⅜ |
| 4½ | 35 | 5⅛ | 5¾ | 3¼ | 2¼ | 2⅜ | 2⅛ | 2⅞ | 1½ | 2¼ | 2⅜ | 2 | 1⅞ | ⅝ | 2½ |
| 4¾ | 37 | 5⅜ | 6 | 3⅜ | 2⅜ | 2⅜ | 2⅛ | 3 | 1 9/16 | 2⅜ | 2⅜ | 2⅛ | 1 15/16 | ⅝ | 2⅝ |

As a check on the diameter of bolts in strap, it should be seen that the loads on them are not in excess of those given in Table XLIII.

**Eccentric Rods.**—These should be sound forgings of mild steel,—the double-eye at the link end being cut out of the solid forging. The diameter at the link end should be that given by,—

**Rule 173.**

**Diameter of eccentric rod at link end** = ·8 D + ·2 inch.

The diameter at mid-length of rod may be calculated in the same manner as a connecting-rod (Rule 81), but for direct-acting engines of ordinary construction,—(where the length of connecting-rod is not more than 2 × stroke for Naval engines, and 2¼ × stroke for Mercantile engines),—the rods may be made with a straight taper from end to end, and may have the following diameters at the ends next eccentrics:—

**Rule 173a.**

| | |
|---|---|
| **Diameter of eccentric rod at eccentric end . . . . . . .** | **Naval**.........D<br>**Mercantile**...D + ·4 inch, |

where D has the same value as before.

The studs by which the T end of the eccentric rod is secured to the strap should be of the same strength as the bolts for valve-rod caps (Rule 169), and for connecting the two parts of the strap (Rule 172$m$), and their diameter may be calculated in the same manner.

The cap-bolts at link end of rod should also, collectively, have the same strength as the valve-rod cap-bolts; but, as they will be considerably smaller, they must not carry so great a load per square inch of section at bottom of thread, and their diameter will be easily fixed by reference to Table XLIII.

## JOY'S VALVE GEAR.

The general arrangement of one form of this gear is shown in Fig. 39. The method of laying down the centre lines is as follows:—

Take a point $d$ (Fig. 40, page 197) on or near the axis of the connecting-rod, such that its extreme transverse vibration is about twice the full stroke of the valve (better rather more than less); and through its extreme positions $d_1$ $d_2$ draw $zz$. Then mark off $e$ and $e_1$,—the extreme longitudinal positions of the point $d$,—and from these points draw two lines to meet in a point $f$ on $zz$, the position of which is such that the angle $efe_1$ is not more than 90° (if there is room a less angle is better).

The point $f$ should be controlled so as to move as nearly as possible on the line $zz$; this may be effected either by an "anchor" link 2,—which may be placed on either side of $zz$,—or by a sliding guide.

Next, on the valve-rod centre 5, mark off $g_1$ and $g_2$, so that $gg_1$ is equal to lap + lead for the top or back end of cylinder, and $gg_2$ equal to same for crank end of cylinder. Then take a point $j$ on the link $e_1f$ so that $e_1j$ is about $3 \times dd_1$, and draw $jg_1$, cutting $zz$ in $m$; $m$ will be the fulcrum of the lever 3, and will also be the centre of the curved reversing block in which the fulcrum slides. When swing links are used (as in Fig. 39), in lieu of a curved block, the arcs

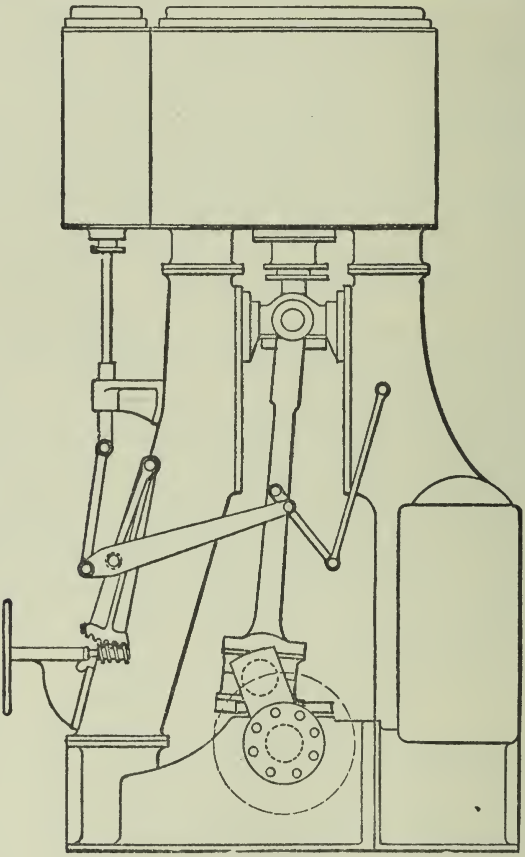


FIG. 39.

described by the link ends, when in ahead and astern positions, must intersect in $M$.

The position assumed for $j$ in the first instance is approximate only, and may not be quite correct; its position must be such that the point $m$ in the lever $jmg_1$ moves to an equal distance on either side of the centre of the reversing block,—also marked $m$. If not at first quite correct, it will generally be obtained on a second trial.

When the gear is correctly set out, the act of moving it from ahead to astern gear, or to any intermediate position, should leave the lead unaltered.

The valve-rod link, which is attached at *g*, may have any convenient length, but the length fixed on determines the radius of the curved guides in the reversing block, or the length of the swing links, whichever construction is adopted.

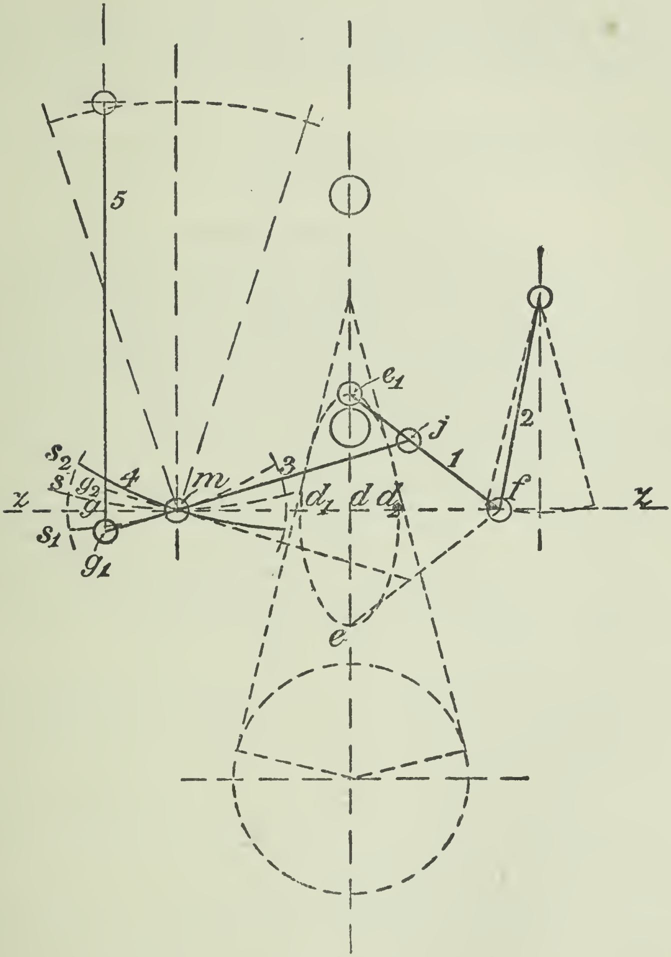


FIG. 40.

It will be noticed that the lap and lead depend on the ratio of *jm* to $mg_1$, whilst the port openings (over and above the leads), depend only on the angle to which the reversing block is canted, and will be about $^4/_5$ths of the distances $ss_1$ and $ss_2$ respectively.

Considerable deviations from the above-described positions and proportions may be made without materially affecting the nature of the results.

Where the distance from centre of piston-rod to centre of valve-rod is different for the different cylinders (H.P., M.P., L.P.), the valve-rod link may be inclined as much as 1 in 12, but beyond that the point $m$ should be moved, and the lengths $jm$ and $mg_1$ altered.

For complete investigation of Bremme's or Marshall's gear see Mr Seaton's "Manual of Marine Engineering."

## REVERSING GEAR.

For all ordinary types of engine, when size renders hand-gear too slow in its action, the simplest and most efficient form of reversing engine is the direct-acting steam cylinder, controlled by a cataract brake. A very perfect form of this gear has long been associated with the name of Messrs Brown of Edinburgh; by the adoption of valve-gear of the "hunting" type they have produced a machine which is handled with the greatest ease and certainty, and which is, at the same time, quite automatic in its movements. The action of the gear is as follows:—On moving the hand-lever through any fraction of its travel, the steam valve is displaced, or opened, a corresponding amount causing the piston to move; likewise as the gear begins to move, the steam valve begins to return to its central or closed position; consequently when the piston has travelled through the fraction of its stroke corresponding with the original movement of the hand-lever, the valve is again central, and the gear at rest.

Direct steam gears should never be fitted without the brake cylinder, as, in careless hands, the steam-gear is apt to "take charge," overpower the controlling hand-gear, and "carry away" some of the brackets, &c.

Another very excellent prime mover, which is adaptable to almost any type of engine, is that known as the "all-round" gear. In this arrangement a connecting-rod from the main lever on the weigh-shaft takes hold of a crank-pin on the side of a worm-wheel, whose worm is revolved by a small reciprocating steam engine.

The great advantage of this gear is that it has no "stops," but can travel round continuously, without risk of carrying anything away, and is therefore perfectly safe in even the most careless hands. It has also the further advantage,—resulting from the crank and connecting-rod principle embodied,—of exerting its greatest power at the extreme positions of links, so that it readily starts to reverse when required and is easily controlled when notching-up is desired. The small engine is sometimes made with two cylinders and cranks to be automatic, and is also sometimes made reversible; but these are both very doubtful improvements.

In all steam reversing gears, arrangements should be made to prevent the accumulation of water in the cylinders of the small engine, in order that they may be in condition to start instantly when the hand-lever is moved.

**Reversing gear weigh-shafts.**—These shafts must be not only strong enough for their work, but also fairly rigid under it, and capable of standing any bending stresses that may come upon them. The following formula gives diameters in accordance with the best modern practice:—

**Rule 174.**

$$D = \sqrt[4]{\frac{d^2 \times \text{No. of gears} \times L \times l}{8}} + \cdot 25,$$

where D = diameter of weigh-shaft in inches;
$d$ = effective diameter of valve-rod in inches;
$l$ = length of drag-rod lever in inches;
L = length of weigh-shaft in feet.

For triple engines of ordinary proportions the above formula gives a diameter of weigh-shaft about equal to twice the effective diameter of the valve-rods,—slightly larger if both forward and after valve-boxes are "outside," and slightly smaller if both are "inside."

Another, a simple and reliable rule for weigh-shafts, is as follows:—

**Rule 174a.** **Diameter** $= \sqrt{\frac{\text{N.H.P.} \times \text{Revs.}}{F}}$

N.H.P. is that given by Rule 7.

F is for two-crank engines, 1000.
F ,, three- ,, 900.
F ,, four- ,, 800.

The designer must, of course, use his discretion and make suitable modifications where the weigh-shaft bearings cannot be placed close to the various levers, or the steam gear lever close to the middle valve-gear in a three-crank engine, or between the two middle gears in a four-crank one.

In some cases the weigh-shaft may be made tapering to the ends, but, as a rule, it is better to use a practically parallel shaft, and, if weight is of great importance, make it hollow, as is commonly the case of large Naval engines; the necessary stiffness is then maintained without difficulty, and there is no fear of the bending.

Large weigh-shafts should be made in two pieces connected by solid flange couplings, and the main lever may be placed between the coupling flanges and secured by passing the coupling bolts through it. "Enlargements" should be provided on the shafts for such other levers as must be fixed at any distance from the ends.

**Size of reversing engine cylinders.** — For ordinary triple engines as described above, this may be obtained from the general relation,—

The steam cylinder of the direct "push and pull" gears should be sufficiently large to act freely with steam at a pressure considerably lower than "boiler pressure."

Rule 175.—Capacity of cylinder should be the capacity of L.P. cylinder ÷ 96.

For all-round gearing with the usual proportion of worm and wheel and length of levers.

Rule 176.—Capacity of a single cylinder should be practically that of the L.P. cylinder ÷ 600.

In their fear of giving too little power in steam reversing gears, there is little doubt that engineers often run to the other extreme, and give much more than necessary, even when two-thirds boiler pressure is assumed. The values assumed for the constants in the above formulæ give ample size.

With "all-round" gears the small engine should not make less than 15 revolutions in moving the gear over, and where possible 20 should be allowed. and up to 25 in large engines.

The pitch of the worm-wheel teeth should never be less than 1½ inches, and for large engines 2½ inches is usual; they should be machine-cut, and should have no more clearance than is absolutely necessary.

The worm is usually forged solid with its spindle, and may have a diameter at pitch given by,—

**Rule 177.**

**Diameter of worm at pitch circle may be . 3 × pitch of teeth.**

**Rule 177a.**

**Width of face of worm-wheel ,, . 2 × ,,**

**Rule 177b.**

**Length of worm (actual thread) ,, . 3 × ,,**

## STEAM TURNING GEARS.

The turning gear now universal is the double worm arrangement,—in which the second, or auxiliary, worm shaft is coupled direct to the crank-shaft of the small turning engine, which makes from 1200 to 2000 revolutions for one turn of the main engines, according to size. The size of cylinder, or cylinders, of turning engine should be such that, with say 50 lbs. of steam, the main engine can be turned one revolution in from 5 to 8 minutes, according to size, which gives,—with the above-mentioned ratios of gearing,—a speed of 250 revolutions per minute for the turning engine.

In the Navy the turning engines have usually two cranks, and are made reversible by means of links or a reversing valve, but one is quite sufficient, and a loose eccentric is all that is required for reversing.

In small vessels, where a special turning engine is not considered necessary, the auxiliary worm-shaft may be driven from a donkey pump or other auxiliary engine, by means of a pitch chain and wheels.

For ordinary three-crank triple and quadruple engines, the sizes of cylinder for turning engine should be such as will satisfy the following relation,—the boiler pressure being assumed as 50 lbs. in all cases:—

**Rule 178.** $$\frac{A \times T}{D^3} = K,$$

where A = area of piston or pistons of turning engine, in sq. inches;
T = travel of piston or pistons, in feet, while turning main engines through one revolution;
D = diameter of crank-shaft of main engines:
K = 14.

**Rule 179.**—In general practice the capacity of the cylinder of a turning engine is = the capacity of the L.P. cylinder of the main engine ÷ 700.

In Naval work the main wheels are commonly of cast-steel with gun-metal rims, but a plain cast-iron wheel, gearing with a hard bronze worm, gives quite as good results and is much cheaper to fit.

However carefully a worm-wheel may be moulded there is sure to be more or less distortion in cooling, and it is therefore the cheapest in the long run to machine-cut the teeth.

The teeth should not be made too fine a fit; there is no objection to a good clearance.

The usual pitch for the teeth of the main wheel is as follows:—

| | |
|---|---|
| For 6-inch to 8-inch crankshaft . . | 2-inch pitch. |
| 8½ ,, 11 ,, ,, . . | 2¼ ,, |
| 11½ ,, 14 ,, ,, . . | 2½ ,, |
| 14½ ,, 18 ,, ,, . . | 3 ,, |
| above 18 ,, ,, . . | 3½ ,, |

When calculating the strength of the teeth, the number in gear at one time should not be taken at more than two.

**Rule 180.**—The length of tooth should not exceed ·6 × pitch, viz. ·26 × pitch outside the pitch circle, and ·34 × pitch within.

It is scarcely necessary to say that the pitch should be an even quarter or half inch, and that the diameter of the wheel must then be made such as will give the required number of teeth; also when the wheel is made in halves the number of teeth should be even.

The pitch of the auxiliary worm-wheel is usually 1½ inches to 2¼ inches.

**Rule 181.**—The width of face of the wheels may be 1·75 × pitch of teeth when a solid worm is used, and 2·1 × pitch when the worm is loose on the spindle.

The diameter of the pitch cylinder of the worm may be,—

**Rule 182.**

**When worm is forged with spindle** . . 2·4 × pitch of teeth.

**Rule 182a.**

**When worm is loose on spindle** { forged steel . 3 × ,,
bronze . 3·3 × ,,

**Length of worm** (actual thread) may be 3 × pitch of teeth.

Wheels up to 3 feet diameter are best made with a continuous plate, or disc, in lieu of arms; larger wheels may have six radial ribs on each side of the disc (the flanges counting as two, when the wheel is in halves), and the disc may then be lightened by cutting six circular, or nearly circular, holes in it.

The thickness of rim, of disc plate, and of ribs, may be same as thickness of tooth at pitch line.

Where possible the main wheel should be placed on a coupling. When cast-steel brackets or bearings are used for the worm-shafts they should be bushed with bronze or white metal.

## SCREW PROPELLERS.

Ships may have any number of screw propellers, and in the past some have had as many as six. In practice the following holds good:—

(i) **Single screw** at the stern for all small ships of sufficient draught of water—all cargo steamers of moderate speed.

(ii) **Two screws,** generally side by side, and called *twin*, for all fast steamers, Naval ships, and for cargo steamers of light draught or high speed.

Two screws, one at the bow and one at the stern, have been tried with success in tugs and other high-powered, small, short craft which are required to maintain full power in a sea-way, inasmuch as when pitching one screw is always fairly immersed.

(iii) **Three screws** have been employed with reciprocating engines in ships of large power and moderate draught—chiefly in Naval ships of foreign powers. Since the introduction of the turbine three screws have been common in ships of all sizes as convenient to the turbine installations and conditions of high revolution with direct drive.

(iv) **Four screws** were originally tried on light-draught ferry-steamers required to move with either end first. In the Navy and Atlantic Service ships of very large size and high speed propelled by turbines have four screws. In the latter case the screws are in pairs, the inner pair being just forward of the rudder, as in twin-screw ships; the outer pair are one on each side, well forward and clear of the feed to the others, and near enough to the skin of the ship to permit of a short outer fin to carry the screw shaft.

*The single screw* is of course the least expensive, but it is not so safe as the twin screws, one of which will propel the ship when the other is out of action, and with both in action she can be steered in case of disablement of the rudder or its gear.

It is better to have a small screw well immersed than a large one with its tips near the sea surface, consequently when the draught of water of a ship is limited and not much more than the diameter of the single screw, two smaller screws will give a higher propulsive efficiency; and in the case of very high power three well-immersed screws will be better than two larger ones insufficiently submerged. When the draught of water is so light that no screw of reasonable diameter can be fitted submerged, then for good efficiency the screw race must be covered in so that when under way the screw submerges itself. This has been done most successfully by Sir J. Thornycroft and Mr Yarrow, the latter using a hinged flap which can be lowered and come into action when the ship is light aft; and when loaded deep enough not to require it the flap is raised out of water.

Screw propellers may have any number of blades; from one to six have been experimented with in actual practice. One blade is theoretically highly efficient; in practice it is objectionable for obvious reasons, chiefly for being out of balance, and its centre of effort revolves and causes complications.

**Two blades** balance, but their centre of effort has an orbit due to the difference of immersion of the blades during revolution, with the usual consequence of considerable vibration. With deep immersion the evil is not so great, and in smooth water the two-bladed screw has a high efficiency. Such screws are now only fitted to small craft, to boats driven by irreversible engines, and to sea-going ships which have to do a considerable amount of sailing. In the latter case the screw is either raised out of water or its blades moved into a fore and aft position, so as to offer little "head" resistance, by such mechanical means as employed to alter the pitch or reverse the blades' angles.

**Three blades** is the commonest for twin- and multiple-screw ships, and when well immersed is the most efficient for all sea-going ships. With the small diameter permissible with direct-driven turbines, there was difficulty in designing a screw blade with sufficient surface with only three, consequently four had to be resorted to for large-powered ships. With geared turbines there is no difficulty.

**Four blades** is the common practice for all single-screw ships, for large merchant ships of full power engaged in Atlantic service, and for twin-screw warships of large power and comparatively high draught, as well as for those other warships and express steamers with three and four screws of small diameter and large surface.

**The shape of the screw blade** is now generally an approximation to an ellipse, but in the mercantile marine there is still a tendency to what was called pear-shape—really an ellipse with truncated end—that is, square tipped. With the small diameter large surface screw it is much broader at the tip and shaped like a Japanese fan; and often the blade is circular, or circular with the tips cut square. Experience has led engineers into the fashion of having the maximum breadth of blade in the middle of small diameter screws, and at about two-thirds of the length from the tip with ordinary ones. In a general way the maximum breadth may be at a distance equal to a quarter of the diameter from the axis.

**The section of the blade** was generally such that the acting face is a true helical surface and the back curved so that any section flattened would be a segment of a circle. Experiments on model screws of a most exhaustive character by Professor D. W. Taylor and Capt. Dyson, U.S. Navy, have proved that while these are good sections they are by no means the best; that on the whole a more ship shape (horizontal or water-line) section is the best, so that the maximum thickness is much nearer the leading than the following edge, and the entering angle more obtuse than was formerly thought good.

**The material of propellers** is a much more important matter now than formerly, as the high rate of revolution obtaining, especially with turbines, sets up centrifugal forces of great magnitude, and the erosion due to the action of sea-water on the surface is severe. Hence it is now necessary in express steamers to have the screw blades made of material which has a very high tensile resistance and a surface which will not easily corrode and permit of erosion.

*Cast-iron* is still used largely in the mercantile marine, and for cargo ships and those working at a moderate rate of revolution is quite satisfactory inasmuch as it is cheap, is strong enough, and will, bar accidents, last six years with common care. If it receives a heavy blow a portion of the blade is broken clean off, leaving the screw to revolve freely, so that when back in port the propeller may be removed and have a new piece "burnt" on; or, in case of a loose-bladed screw, a new blade can be shipped.

*Bronze of kinds*, mostly high tensile zinc and copper alloys with some "doctor" metal added, are in general use for express steamers, Naval ships, and other craft where cost is not so important a factor as high efficiency. The bronze screw is always thinner than cast-iron, has a smoother surface, and is cast truer to pitch, and wastes very little under the action of sea-water; moreover, it is much lighter to handle and carry. Phosphor bronze has also been used for blades, and from its high elastic limit and smoother surface has advantages, but it is not so tough.

*Naval bronze or gun-metal* is now seldom, if at all, used, as it is more costly and has a much lower elastic limit.

*Steel castings* were at one time used for large ocean-going express steamers; blades of this material were generally cast separate and had a very rough surface, which was not improved by the action of sea-water; they were seldom true to pitch and often very badly out from tip to root. Corrosion was rapid, so that their working life under favourable circumstances was only four years, and all attempts to prevent destruction proved failures.

**The dimensions of a screw propeller** depend on the circumstances of each case. Sometimes they have to conform to the requirements of the engines, and sometimes the engines must be designed to suit the screws. Oftener still they are the result of compromise between the demands of both.

*The diameter of a screw* is governed by the draught of water of the ship and the form of the stern lines, the exceptions to the latter have been already cited. The screw should have its top blade immersed to the extent of 15 per cent. of its diameter to be efficient on ocean service, and 10 per cent. for smooth water work. A ship having fine water-lines in the "run" will permit of a better "feed" to the screw and have wake currents less extensive than one with a full "run," consequently she may have a screw of smaller diameter. In the same way a twin-screw ship may have smaller screws *pari passu* than a single-screw ship of the same form. All other things being equal, the power delivered to the screw per revolution, that is $a \frac{\text{I. H. P.}}{\text{R}}$ is the criterion of dimensions. The screw has with that power to project a column of water, so that by its reaction an adequate thrust on the ship is produced; within reasonable limits it may be taken that if D be the diameter of the column, and L its length, then

$$\frac{\pi D^2}{4} \times L = \text{constant}; \text{ that is } D^2 \times L \text{ is constant.}$$

In other words, that $D^2$ varies inversely as L, so that if the screw's diameter increases the pitch may be less and *vice versa.* Consequently for a given power there may be a great variety of dimensions for the screw. There may be one with small diameter and big pitch, or one of large diameter and smaller pitch.

**Pitch ratio** is the expression for the relation of pitch to diameter, and may be anything the designer chooses, but practice has shown that $\frac{P}{D}$ should not be much less than 1·0, and propulsive results are better when it is greater than 1, and the best are attained with ordinary ships when the pitch ratio is from 1·2 to 1·6.

**The slip of a screw** is of course a real thing as well as an apparent one. That is, the speed of the ship through the water is less than the multiple of the revolutions and acting pitch, and the difference between them is the *apparent slip. The real slip* is the movement of the water imparted to it by the propeller through the surrounding water, and what that is cannot be measured; in any and every case it is more than the apparent, and with a bluff ship with enormous wake currents it is much more, and of course is always positive. *Negative apparent slip* is often observed with such ships when they have screws of large diameter and surface.

**Surface of Screw** (acting) is that of the blades from the boss to the tip acting on the water as the screw turns; and its area is that actually measured on the blade.

**Projected surface** is the area projected on a plane transverse to the screw's axis from the acting surface, and is often taken by preference as the criterion of the screw capability.

**Thrust.**—If V is the velocity of the stream of water issuing from the ship, whose own velocity is $v$ in feet per second; then velocity of the stream with respect to *still water* past which the ship is moving will be $(V - v)$. This is the *real slip.* Then:—

**Rule 183.** Real slip per cent. of velocity of flow $= \frac{(V-v)}{V}100.$

**Rule 184.** The mass of sea-water will be

$$\frac{\pi D^2}{4}\times V\times\frac{64}{32}=1{\cdot}571\ D^2\times V.$$

The momentum of the stream $= 1{\cdot}571\ D^2V(V-v) =$ thrust in lbs.

The work done $= 1{\cdot}571\ D^2V(V-v)v$ in foot-lbs.

**Rule 185.** Horse-Power $= \frac{1{\cdot}571\ D^2V(V-v)v\times 60}{33{,}000} = \frac{D^2V(V-v)v}{350}.$

The diameter of the screw may be determined by this formula as follows:—P is the pitch, R the revolutions per minute; $\frac{V-v}{v}=f$, a fraction of V; E is the efficiency of the screw and engine, that is the ratio of actual work of propulsion to the I.H.P. developed by the engine; or in the case of turbines is the ratio to the S.H.P., and therefore the efficiency of the screw only. Then the following holds good,

$$\text{I.H.P.}=\frac{D^2\times(P\times R)^3(f-f^2)}{350\ E}.$$

**Rule 186.** Diameter of Screw in feet $= \sqrt{\frac{\text{I.H.P}\times 350\ E}{(P\times R)^3\times(f-f^2)}}.$

If the real slip be taken as $x=0{\cdot}2P_c+f$ for twin screws and $0{\cdot}18\ P_c+f$ for single and centre screws of three-screw ships, where $f$ is as before and $P_c$ is the prismatic co-efficient of the ship's submerged hull, then

**Rule 186a.** Diameter of Screw in feet $= \sqrt{\frac{\text{I.H.P.}}{x(1-x)}\times\left(\frac{450}{P\times R}\right)^3}.$

A good practical method of determining the diameter of a screw suitable for a particular ship is by means of

**Rule 186b.** Diameter of Screw in feet $= z\times P_c\ \sqrt[3]{\frac{\text{I.H.P.}}{R}}$

| | | | |
|---|---|---|---|
| For single screws ordinary, | $z=7{\cdot}25$. | Ocean-going express, | 7·61. |
| ,, twin ,, | $z=6{\cdot}55$. | ,, ,, | 6·88. |
| ,, quadruple ,, | $z=6{\cdot}25$. | ,, ,, | 6·51. |
| ,, turbine driven centre, | $z=6{\cdot}55$. | ,, ,, | 6·88. |
| ,, ,, wing, | $z=5{\cdot}75$. | ,, ,, | 6·04. |

A rough and ready method suitable to cargo steamers of good draught.

**Rule 186c.** Diameter of Screw in feet $= z\sqrt{D\times S}$

D is the diameter and S the stroke of L.P. piston in feet.
$z=(2{\cdot}4+P_c)$ for twin screws and $(2{\cdot}7+P_c)$ for single screws.

**Pitch of screw** is now generally uniform throughout, so that the propeller is said to be a *true screw*. If the screw is of small diameter for the power delivered to it, there is of necessity a larger slip ratio than one with larger diameter. Slip ratio is a variable factor, and may be chosen by the designer to suit the other conditions. It is now known, as the result of the very many and carefully made experiments, what the efficiency is of any particular screw at different slip factors; or with the same slip factor the efficiency with different pitch ratios. Curves of efficiency, with these variables, may be set out and a choice made to suit the revolutions of the engines. In the mercantile marine, with reciprocating engines an *apparent slip* of 10 per cent., and with turbines 15 per cent. is considered satisfactory. Bluff cargo boats seldom show so high a ratio, 5 per cent. being usual, and negative slip not unusual with large screws.

If S is the speed of the ship in knots per hour, R the revolutions per minute, and $s$ the slip in knots,

$$\text{Pitch} = \frac{(S+s) \times 6080}{60\ R}.$$

If $x$ is the slip per cent. of speed of screw, that is of $\frac{P \times R \times 60}{6080}$,

then $$S = \text{speed of screw}\left(1 - \frac{x}{100}\right).$$

**Rule 187.** $$\textbf{Then Pitch} = \frac{S}{R} \times \frac{10{,}133}{100 - x} \text{ feet.}$$

By referring to Rule 186 the slip for any set of conditions can be estimated and the pitch of screw calculated accordingly, or the pitch may be assumed from engine conditions and the diameter calculated by that rule with the assumed apparent slip as *f*.

**The surface of the screw** is after all the most difficult to determine, that the very highest propulsive efficiency possible may be obtained with given diameter and pitch. The tendency of any increase in blade area is to add to the resistance to turning, as the decrease is to an augmentation of apparent slip without gain in propulsive efficiency—in other words, the actual column of water is not thereby increased with increase in revolutions. To go to extremes, a propeller of suitable diameter and pitch may fail to propel efficiently, inasmuch as the blade surface is insufficient to hold against the water, the blades being simply whirled through it. Starting with such a screw, any addition to blade surface tends to avoid this plunging through the water, so that the efficiency is improved; each addition makes a further improvement, but not at so great a rate; finally, the further increases produce no improvement, and the last addition detracts from efficiency. Breadth of blade, too, has considerable influence on efficiency of surface, so that with the minimum surface on the fewest blades there is the maximum breadth of blade and the greatest efficiency. The following is in accordance with good practice:—

**Rule 188.** **Area of Acting surface in sq. ft.** $= Q\sqrt{\frac{\text{I.H.P.}}{\text{Revs. p. m.}}}$.

$Q = P_c \times M$ where

| | | | |
|---|---|---|---|
| For four-bladed screws, | single, | M = 20·0. | For twin screws, 15·0. |
| ,, three ,, | ,, | M = 19·0. | ,, ,, 14·3. |
| ,, two ,, | ,, | M = 17·5. | ,, ,, 13·1. |

**Maximum breadth of blade.**

**Rule 189.** **Maximum Breadth in inches** $= K\sqrt{\frac{\text{I.H.P.}}{\text{Revs. per min.}}}$

where K = 14 in four-bladed screws; K = 17 in three-bladed; and K = 22 in two bladed. For turbines K is much higher when the diameter is so limited.

**Thickness of blades** in cases of very high rate of revolution and large power should be calculated most carefully by estimating the tension due to centrifugal force and the tension on the metal near the face due to the bending moment on that section from the reaction producing thrust.

In designing a screw it is not unusual to estimate a certain hypothetical thickness at the axis and get the longitudinal section by drawing a line from the tip to the points on the axis; then taking the thickness at the tip from practical considerations, draw a line parallel to the face line at a distance equal to this thickness till it meets the former line. This hypothetical thickness may be calculated by the following:—

**Rule 190.** **Thickness of blade at axis** $= \sqrt{\frac{d^3}{n \times b}} \times X$.

The value of X for cast-iron is 6·3; for gun-metal (Admiralty bronze) 3·2; for strong bronzes 2·35; and for steel 2·5.

$d$ is the diameter of the tunnel shafting or that required to resist torsion, and may be taken as $\sqrt[3]{\frac{\text{I. H. P.}}{R}} \times H$.

The value of H for triple and quadruple compound engines may be taken as $5·8\sqrt{p}$, and for turbines 65. Substituting

**Rule 191.** **Thickness of blade at axis** $= X\sqrt{\frac{\text{I.H.P.}}{R} \times \frac{H}{n \times b}}$,

where $n$ is the number of blades and $b$ the breadth at the root in inches near the boss, and $p$ the boiler pressure.

But it may be preferred to estimate the thickness of blade at a point where it actually exists and has a definite breadth, say at 1½ diameters of shaft from the axis. In this case the same rule applies, but the value of X is different.

**Rule 191a,** when thickness is estimated at $1·5 \times d$ from the axis, then for cast-iron X = 4; 2 for Admiralty bronze; 1·6 for cast-steel; and 1·5 for forged steel and strong bronzes.

The thickness of blades at the tip should be as follows:—

**Thickness of blade at tip**—Cast-iron, . . $\cdot04D + \cdot4$ inch.
„ „ Cast-steel, . . $\cdot03D + \cdot4$ „
„ „ Gun-metal, . . $\cdot03D + \cdot2$ „
„ „ High-class bronze, . $\cdot02D + \cdot3$ „
where D is diameter of propeller in feet.

The thickness of blade for loose-bladed Naval propellers may be taken roughly by :—

**Rule 191b.**
$$t = \sqrt{\frac{T \times L}{d \times K}} + \cdot25,$$

where $t$ = thickness of blade in inches at surface of boss sphere ;

$$T = \text{indicated thrust per blade} = \frac{\text{I.H.P.} \times 33{,}000}{\text{Pitch} \times \text{revs.} \times \text{No. of blades}};$$

$L = \cdot6^{*} \times$ total length of blade (flange joint to tip) ;
$d$ = diameter of blade flange ;
$$K = \text{co-efficient} = \begin{cases} 800 \text{ for Admiralty bronze;} \\ 950 \text{ for strong bronze.} \end{cases}$$

## Table LXVII.—Thickness of Propeller Blades.

| Dia. of shaft in inches. | Solid cast-iron, 4 blades. | | Loose bladed propellers. | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Breadth of blade = length of boss = 2·7 × dia. of shaft. | Thickness of blade (at centre of shaft.) | Breadth of blade = ·8 × dia. of flange = 1·8 × dia. of shaft. | Thickness of blade at centre of shaft. | | | | |
| | | | | 4 blades cast-iron. | 4 blades steel or bronze. | 3 blades steel or bronze. | 4 blades gun-metal. | 3 blades gun-metal. |
| inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. | inches. |
| 6 | 16 | 4¼ | 10¾ | 5 | 3 | 3½ | 3½ | 3⅞ |
| 7 | 19 | 4¾ | 12½ | 5¾ | 3½ | 3⅞ | 3⅞ | 4½ |
| 8 | 21½ | 5½ | 14½ | 6½ | 3⅞ | 4½ | 4½ | 5⅛ |
| 9 | 24 | 6 | 16¼ | 7¼ | 4⅜ | 5 | 5 | 5¾ |
| 10 | 27 | 6⅝ | 18 | 8 | 4⅞ | 5½ | 5½ | 6¾ |
| 11 | 30 | 7¼ | 19¾ | 8¾ | 5¼ | 6⅛ | 6⅛ | 7 |
| 12 | 32½ | 7⅞ | 21½ | 9½ | 5¾ | 6⅝ | 6⅝ | 7⅝ |
| 13 | 35 | 8½ | 23½ | 10¼ | 6¼ | 7⅛ | 7⅛ | 8⅛ |
| 14 | 38 | 9 | 25¼ | 11 | 6⅝ | 7⅝ | 7⅝ | 8¾ |
| 15 | 40½ | 9⅝ | 27 | 11¾ | 7⅛ | 8¼ | 8¼ | 9⅜ |
| 16 | 43 | 10¼ | 28¾ | 12½ | 7⅝ | 8¾ | 8¾ | 10 |
| 17 | 46 | 10⅞ | 30½ | 13¼ | 8 | 9¼ | 9¼ | 10⅝ |
| 18 | ... | ... | 32½ | 14 | 8½ | 9¾ | 9¾ | 11¼ |
| 19 | ... | ... | 34¼ | ... | 8⅞ | 10¼ | 10¼ | 11¾ |
| 20 | ... | ... | 36 | ... | 9⅜ | 10¾ | 10¾ | 12⅜ |

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* For ordinary Griffiths form of blade ; if blades are broad at tip this quantity should be increased, and *vice versa*.

If cast-iron of lower than Admiralty quality (tensile strength 9 tons per square inch) be used, suitable increase in thickness of blade must be made.

If breadth of blade be made less than length of boss, thickness must of course be so increased that value of b × $t^2$ shall always agree with that found from above Table.

A good method of setting out the longitudinal section of the blade is as follows :—

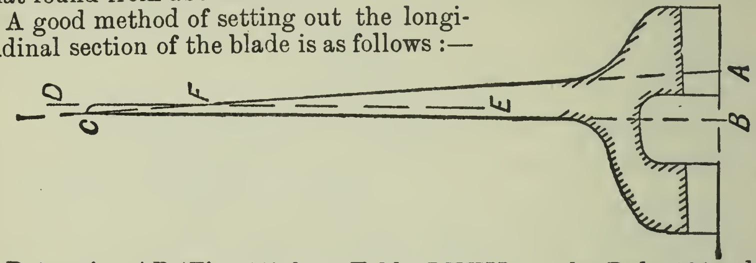

FIG. 41.

Determine AB (Fig. 41) from Table LXVII., or by Rule 191 ; draw AC so as to give no thickness at tip ; set off thickness at tip (from Table LXVIII.), and draw DE parallel to CB (at right angles to shaft axis), and then fill in the back of the blade near F so as to make DF die gradually into AC.

### Table LXVIII.—Thickness of Propeller Blades at Tip.

| Diameter of Propeller in feet. | Thickness at tip, in inches. | | |
|---|---|---|---|
| | Cast-iron ·04D+·4 | Cast-steel. ·03D+·4 | Bronze ·03D+·2 |
| 7 | 11/16 | 5/8 | 7/16 |
| 8 | 3/4 | 5/8 | 7/16 |
| 9 | 3/4 | 11/16 | 7/16 |
| 10 | 13/16 | 11/16 | 1/2 |
| 11 | 13/16 | 3/4 | 1/2 |
| 12 | 7/8 | 3/4 | 9/16 |
| 13 | 15/16 | 13/16 | 9/16 |
| 14 | 15/16 | 13/16 | 5/8 |
| 15 | 1 | 7/8 | 5/8 |
| 16 | 1 1/16 | 7/8 | 11/16 |
| 17 | 1 1/16 | 15/16 | 11/16 |
| 18 | 1 1/8 | 15/16 | 3/4 |
| 19 | 1 3/16 | 1 | 3/4 |
| 20 | 1 3/16 | 1 | 13/16 |
| 21 | 1 1/4 | 1 1/16 | 13/16 |
| 22 | 1 1/4 | 1 1/16 | 7/8 |
| 23 | 1 5/16 | 1 1/8 | 7/8 |
| 24 | 1 3/8 | 1 1/8 | 15/16 |

**Studs or screws for attaching blades to boss.**—These are usually of one of the stronger bronzes, for bronze propellers, and of mild steel for cast-iron or cast-steel propellers. They should be of size given by following formula :—

**Rule 192.**

$$a \times N \times r = \frac{T \times L}{K}$$

where $a$ = area of one stud or screw at bottom of thread (sq. inches);

$N$ = number of studs or screws for one blade (usually 7 to 11);

$r$ = radius of studs or screws (in inches);

$T$ = indicated thrust per blade $= \frac{\text{I.H.P.} \times 33{,}000}{\text{Pitch} \times \text{revs.} \times \text{No. of blades}}$;

$L = \cdot 6 \times$ total length of blade (flange joint to tip) in inches;

$K = \begin{cases} 1700 \text{ for studs, steel and strong bronzes ;} \\ 1400 \text{ for Naval bronze bolts.} \end{cases}$

For ships constantly running in rough waters or in crossing the Atlantic, these factors should be 20 to 25 per cent. larger. With cargo steamers, which have often to steam when light at good power, and all of it taken by one blade, there is need of the same modification. On the other hand, ships in smooth water service, or only occasionally exerting full power, may have much lighter blades, so that K in that case may be 10 per cent. less without risk.

**The weight of propellers** may be estimated with a good approximation to the actual by the following :—

**Rule 192a.**

$$\textbf{Weight in cwt.} = \frac{\text{Surface in sq. ft.} \times \text{thickness of root in inches}}{Y}.$$

The value of Y for solid cast-iron screws is 4·5.
,, ,, ,, bronze ,, 3·8.
,, ,, ,, steel ,, 4·0.
The value of Y for loose-bladed cast-iron screws is 3·0.
,, ,, ,, cast-steel ,, 2·8.
,, ,, ,, bronze ,, 2·5.

**Propeller Bosses** when cast separately from the blades are often of the same material, but as the stresses when fitting on the shaft and by the action of the blades when at work are often severe, they are better of a somewhat tougher metal. Thus cast-iron bosses should be of a tough mixture, of which hematite or even steel is a component part; when weight is not of great moment such tough cast-iron bosses of ample thickness can be used with bronze and steel blades, and steel bosses, which can be made now quite cheaply, do very well with bronze blades, and are of course much cheaper than bronze ones.

The bosses of solid propellers are usually egg-shaped; those for loose blades are spherical, and sometimes with a tapering tail to avoid cavity at their rear at high speed of ship.

**Rule 193.**

$$\textbf{The diameter of boss for loose blades} = 0 \cdot 9 \sqrt{\text{diameter of screw}}.$$

The length of boss for three and four blades is usually 0·75 its diameter.

The bosses of solid propellers are usually in diameter about ⅙ to ⅛ the diameter of the screw and 2·5 to 2·75 diameter of shaft in length.

**The bolts or studs for securing blades** are usually of strong bronze, but sometimes studs are of steel with bronze cap nuts.

The tension on them due to centrifugal force $= \dfrac{Wv^2}{32r}$.

W is the weight of a blade in lbs., $v$ the velocity in ft. per sec. at centre of gravity of blade, whose distance from the axis is $r$ feet.

If R be the revolutions per min. and the pitch P, then

$$v = \frac{R}{60}\sqrt{P^2 + (2\pi r)^2}.$$

The diameter of blade bolts whose number is $n$ may be calculated from these rules and found as in practice by the following :—

**Rule 194. Diameter of blade bolts** $= \dfrac{d \times z}{n}$,

where $z = 1{\cdot}6$ for a three-bladed screw and $1{\cdot}3$ for a four-bladed, $d$ being as before the equivalent diameter of shaft for torque.

**Blade flanges.**—These should be of the following proportions :—

**Rule 195. Diameter of flange = 2·25 × diameter of tail shaft.**

**Rule 195a.**

$$\textbf{Thickness of flange} = \begin{cases} {\cdot}85 \times \text{diam. of stud for G.M., bronze, or steel.} \\ 1{\cdot}4 \times \quad ,, \qquad ,, \ \text{cast-iron.} \end{cases}$$

**Proportions of propeller boss.**—The bosses of solid cast-iron propellers may also be of the following proportions :—

**Rule 196. Length of boss = 2·7 × diameter of tail shaft.**

**Rule 196a. Diameter of boss = 2·7 × ,, ,,**

**Rule 196b.** The fore and aft section of the boss should be oval,—the principal radius being ·8 × diameter of boss.

The length of the tapered bore may be divided into three approximately equal parts, of which the two end ones will bear on the shaft, while the central one will be cored back.

The thickness of "shell" of the central, or cored out, portion of the boss should be,—

**Rule 196c. Thickness of boss = ·65 × thickness of blade at shaft axis.**

Where the blade flanges are all within the sphere of the boss,—as is usual in Naval propellers,—the extreme diameter of boss is about $0{\cdot}9\sqrt{\text{diameter of propeller}}$; the length of such a boss should be the same as that of a cast-iron boss (Rule 196).

As previously mentioned under "Tail shafts," the taper to which the boss is bored out should be about 1 inch on the diameter for each foot of length, but never less than ¾ inch.

## Table LXIX.—Particulars of Screw Propellers in various Ships.

| Reference. | Description of ship. | | | | Results of trials. | | | | Particulars of screw. | | | | Ratios of | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | Speed knots. | Revs. pr. min. | I.H.P. / Revs. | $D.^{2/3}S^3$ / I.H.P. | Diamr. ft. | Pitch ft. | No. of Blades. | Surface sq. ft. | Pitch / Diamr. | Surface / $\pi r^2$ | Slip pr. cent. |
| Mr | Single | Screw | Passenger | S | 15·42 | 76·2 | 55·52 | 245 | 19·00 | 21·00 | 4 | 100 | 1·105 | 0·353 | 2·38 |
| Oo | Do. | do. | do. | S | 16·16 | 64·6 | 86·66 | 245 | 20·00 | 29·00 | 4 | 100 | 1·450 | 0·318 | 12·60 |
| Wdg | Twin | do. | Express | S | 22·80 | 77·2 | 194·30 | 281 | 22·31 | 32·83 | 4 | ... | 1·471 | ... | 8·89 |
| Ffs | Single | do. | Cargo | S | 11·39 | 69·9 | 52·00 | 251 | 19·25 | 19·25 | 4 | 120 | 1·000 | 0·412 | 14·30 |
| Fso | Do. | do. | Passenger | S | 14·62 | 79·75 | 71·40 | 246 | 19·50 | 21·00 | 4 | 118 | 1·077 | 0·394 | 11·65 |
| Dn | Do. | do. | do. | S | 16·82 | 72·00 | 64·72 | 270 | 19·50 | 26·50 | 4 | 115 | 1·359 | 0·385 | 10·70 |
| Ao | Do. | do. | do. | S | 14·85 | 79·00 | 32·91 | 220 | 16·00 | 20·00 | 4 | 85·0 | 1·250 | 0·423 | 4·8 |
| Tns | Do. | do. | Cargo | S | 11·27 | 69·00 | 28·03 | 275 | 16·33 | 18·50 | 4 | 82·0 | 1·133 | 0·391 | 10·55 |
| Fos | Twin | do. | Passenger | S | 16·60 | 89·00 | 58·18 | 275 | 18·75 | 20·5 | 4 | 84·0 | 1·093 | 0·304 | 7·80 |
| Fsf | Do. | do. | do. | S | 17·33 | 89·50 | 42·15 | 258 | 16·50 | 22·0 | 4 | 69·0 | 1·333 | 0·322 | 10·90 |
| Fsn | Do. | do. | Cargo | S | 13·56 | 83·50 | 35·21 | 296 | 16·75 | 18·25 | 4 | 78·0 | 1·090 | 0·355 | 13·56 |
| Fns | Do. | do. | do. | S | 12·88 | 83·00 | 27·32 | 220 | 15·50 | 17·50 | 4 | 61·5 | 1·130 | 0·324 | 10·25 |
| Fso | Do. | do. | Passenger | S | 15·09 | 83·60 | 25·06 | 239 | 14·50 | 20·00 | 4 | 58·0 | 1·380 | 0·351 | 15·09 |
| By | Single | do. | Cargo | S | 13·30 | 83·0 | 18·69 | 206 | 14·25 | 18·00 | 4 | 63·0 | 1·263 | 0·396 | 6·80 |
| Lr | Do. | do. | do. | S | 13·23 | 91·5 | 17·32 | 235 | 13·75 | 17·00 | 4 | 57·0 | 1·236 | 0·385 | 13·80 |
| Eo | Do. | do. | Passenger | S | 14·80 | 88·0 | 23·66 | 192 | 13·50 | 21·00 | 4 | 67·0 | 1·414 | 0·469 | 18·80 |
| Va | Do. | do. | do. | S | 14·90 | 106·0 | 14·80 | 224 | 12·00 | 16·00 | 4 | 48·0 | 1·333 | 0·425 | 11·00 |
| Jo | Do. | do. | do. | S | 13·23 | 101·0 | 11·81 | 205 | 11·50 | 14·50 | 4 | 50·0 | 1·261 | 0·481 | 8·00 |
| Ba | Do. | do. | Cargo | S | 10·80 | 87·0 | 7·60 | 228 | 10·50 | 15·75 | 4 | 36·5 | 1·500 | 0·420 | 15·00 |
| Po | Do. | do. | do. | S | 11·30 | 98·0 | 5·05 | 178 | 9·50 | 12·75 | 4 | 34·5 | 1·342 | 0·486 | 8·30 |
| Qn | Do. | do. | do | S | 11·07 | 132·0 | 2·46 | 168 | 7·75 | 9·25 | 4 | 25·0 | 1·194 | 0·530 | 8·13 |
| Mpl | Twin | do. | Tug . | . | 11·10 | 141·0 | 1·51 | 123 | 6·50 | 10·75 | 3 | 17·0 | 1·654 | 0·515 | 25·70 |
| Gtn | Single | do. | Yacht | . | 9·96 | 217·0 | 0·417 | 154 | 4·50 | 6·75 | 3 | 5·5 | 1·500 | 0·344 | 22·0 |

## Table LXIXa.—Particulars of Bronze Screw Propellers (High Speed).

| Reference. | Description of ship. | Results of trials. | | | | Particulars of screw. | | | | Ratios of | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Speed knots. | Revs. pr. min. | I.H.P. / Revs. | $D^{2/3} S^3$ / I.H.P. | Diamr. ft. | Pitch ft. | No. of Blades. | Surface. sq. ft. | Pitch / Diamr. | Surface. / $\pi r^2$ | Slip pr. cent. |
| Dk | First Class Cruiser | 23·05 | 116·0 | 131·7 | 234 | 19·0 | 24·5 | 3 | 76·0 | 1·290 | 0·269 | 17·8 |
| D2 | Do. do. do. | 24·11 | 122·4 | 127·5 | 262 | 19·0 | 23·0 | 3 | 105 | 1·210 | 0·370 | 13·2 |
| Wr | Do. do. do. | 23·28 | 139·4 | 87·5 | 298 | 15·75 | 20·7 | 3 | 85·0 | 1·317 | 0·436 | 15·7 |
| En | Do. do. do. | 20·05 | 99·4 | 53·6 | 276 | 16·50 | 23·3 | 3 | 60·0 | 1·412 | 0·280 | 12·3 |
| Ln | Battleship | 18·60 | 110·0 | 72·70 | 246 | 17·50 | 19·75 | 4 | 86·0 | 1·129 | 0·354 | 13·3 |
| Cg | 2nd Class Cruiser | 21·03 | 160·0 | 40·62 | 233 | 12·25 | 15·00 | 3 | 46·0 | 1·225 | 0·389 | 11·2 |
| Fs | Do. do. do. | 20·10 | 140·8 | 36·47 | 254 | 14·50 | 16·50 | 3 | 44·1 | 1·138 | 0·267 | 12·3 |
| Fd | Scout | 25·29 | 211·8 | 34·25 | 213 | 12·00 | 14·50 | 3 | 57·6 | 1·208 | 0·509 | 16·6 |
| Sr | 3rd Class Cruiser | 22·45 | 253·3 | 20·16 | 230 | 11·00 | 10·50 | 3 | 43·0 | 0·955 | 0·453 | 14·5 |
| Dv | Destroyer | 30·07 | 385·7 | 7·00 | 239 | 6·75 | 9·00 | 3 | 16·0 | 1·332 | 0·447 | 12·2 |
| Sn | Do. | 27·60 | 355·0 | 5·01 | 237 | 6·60 | 8·75 | 3 | 12·75 | 1·326 | 0·370 | 10·0 |
| La | Turbine Express Str. | 26·46 | 191·5 | 94·5 | 268 | 16·5 | 16·13 | 4 | 110·0 | 0·979 | 0·514 | 13·2 |
| Ca | Do. do. Str. | 20·19 | 185·0 | 43·2 | 312 | 14·0 | 13·00 | 4 | ... | 0·929 | ... | 15·0 |
| Bm | Do. do. Str. | 23·27 | 456·0 | 10·74 | 192 | 7·17 | 6·67 | 3 | ... | 0 930 | ... | 22·4 |
| Sm | Do. Cruiser | 25·95 | 378·4 | 25·37 | 256 | 9·50 | 8·70 | 3 | ... | 0·916 | ... | 20·1 |
| At | Do. do. | 23·63 | 450 | 10·44 | 193 | 6·50 | 6·56 | 3 | 19·6 | 0·969 | 0·561 | 20·4 |
| Ns | Do. Scout | 27·13 | 757 | 7·12 | 118 | 5·50 | 5·00 | 3 | 19·1 | 0·909 | 0·804 | 25·7 |
| St | Do. do. | 30·37 | 631 | 10·13 | 179 | 6·60 | 6·30 | 3 | 21·1 | 0·955 | 0·621 | 22·6 |
| Cd | Express Recipros. | 18·10 | 134 | 20·34 | 188 | 12·00 | 16·50 | 3 | 40·5 | 1·375 | 0·358 | 13·6 |
| Ce | Do. do. | 15·03 | 89·5 | 13·26 | 223 | 11·50 | 19·00 | 3 | 34·0 | 1·652 | 0·327 | 15·1 |
| Pg | Do. do. | 20·30 | 177 | 19·80 | 194 | 10·50 | 14·30 | 3 | 36·0 | 1·362 | 0·439 | 18·9 |
| Pe | Do. do. | 18·9 | 189 | 8·50 | 246 | 9·00 | 12·00 | 3 | 26·0 | 1·333 | 0·409 | 15·6 |
| Ze | Twin Screw Yacht | 12·8 | 147 | 3·54 | 165 | 7·75 | 11·00 | 3 | 21·0 | 1·420 | 0·460 | 19·0 |

## THE PADDLE-WHEEL PROPELLER.

**The common radial wheel with fixed floats** is now seldom seen, inasmuch as it is far less efficient especially with limited diameter than one with feathering-floats. It is, however, a much simpler structure and easy of repair with simple means, and can be constructed largely of timber, as it is indeed on the large tug steamers still employed on the big American rivers, with repair shops few and far between.

There are two positions for this propeller, viz. one wheel on each side, or a single one or pair of wheels at the stern of the ship. The latter position is preferred for river service and in narrow channels. The side wheel is still a favourite propeller for tugs in harbour duties, where manœuvring ability and short, smart service is highly desirable and much backing and filling has to be done.

The effective diameter of a radial wheel is somewhat difficult to determine,—as it depends very much on the form of float, the amount of immersion, the waves set up by the wheel, &c.,—but it is usually measured between the centres of the opposite floats.

Taking the feathering wheel as the one in general use, of diameter D to float centres, moving at R revolutions (at full speed) per min.

A is the area of one float in square feet.

V is its velocity in feet per second and therefore $= \pi D \times \frac{R}{60}$.

S the speed of the ship in knots.

$v$ the velocity of the ship in feet per second $= 1{\cdot}689 \times S$.

$f$ the fraction that the slip is of $V = \frac{V - v}{V}$.

$e$ is the efficiency of the engines.

E the efficiency of engines and wheels.

$T_R$ the tow rope resistance of the ship in lbs. at speed S.

The power transmitted to the wheels is then $= \text{I.H.P.} \times e$.

The power delivered by the wheels $= \text{I.H.P.} \times E$.

The stream projected by each wheel is measured by $A \times V \times 64$ lbs.

The mass of such stream is $\frac{A \times V \times 64}{32}$ or $2A \times V$.

The acceleration given is $V - v$. Hence

The pressure of float $= 2A \times V(V - v)$ which is equal the thrust.

The work done $= 2A \times V(V - v)v$ or $120A \times V(V - v)v$ per min., which will equal $T_R \times 60V$ and also $\text{I.H.P.} \times E$.

**Rule 197.** **The diameter of the wheel** $= 19 \times \frac{V}{R}$.

Taking the acceleration as $fV = f \times \frac{\pi D \times R}{60}$.

**Rule 198.** **Then the thrust** $= \frac{Af}{1800}(\pi D \times R)^2$. Hence

**Rule 199.** **Area of float** $A = \frac{\text{Thrust} \times 1800}{f(\pi D \times R)^3} = \frac{\text{Thrust} \times 182}{f(D \times R)^3}$.

Modifying this rule in accordance with good modern practice, the following rule holds good :—

**Rule 199a.** **Area of one feathering float** $= \frac{\text{I.H.P.}}{f - f^2} \times \left(\frac{C}{D \times R}\right)^3$.

If I.H.P. is the gross amount developed and E is $= 0 \cdot 6$, then—

Value of C is 83·2 for a pair of wheels.
Value of C is 109 for a single stern wheel.

In some modern high-class swift steamers E is sometimes as high as 0·66 when clean and in smooth water, in such cases C may be 85·6. On the other hand, for tugs and small ocean-going craft it will be less than 83·2.

The apparent slip of a paddle wheel is larger with a radial than a feathering wheel, and the amount will depend on the circumstances of each case ; where a small "race" only is possible its velocity—that is, its slip—must be greater than if the section were larger. In shallow draught steamers, and on side wheelers for narrow passages through locks or dock entrances, the area of float is necessarily circumscribed—consequently for high power the acceleration must be great. In practice the rate of slip is from 15 to 30 per cent. with radial floats, and 15 to 20 with feathering ; in some cases, due to circumstances above-named, it is even with the latter 25 per cent. at full speed. Generally with a reduction in power and speed of the ship there is a reduction in the rate of slip.

The wheel race of a paddle steamer is not affected nearly so much by hull currents as is the screw, except of course with stern wheels, when wake currents and augmented resistance play a prominent part in propulsive efficiency.

The number of floats on a paddle wheel varies with the diameter.

**Rule 200.** **Radial fixed floats, one for each foot of diameter.**

**Rule 200a.** **Number of feathering floats** $= \frac{D + 2}{2}$ or $\frac{60}{\sqrt{R}}$.

**Proportion of Paddle Floats.**—A the area, B the breadth ; $r$ the ratio of length to breadth, so that $A = B \times rB = rB^2$ ; then

**Rule 201.** **Breadth of float** $= \sqrt{\frac{A}{r}}$.

**Rule 201a.** **Length of float** $= r\sqrt{\frac{A}{r}}$.

In practice $r$ is 4 to 5 with radial wheels and 2·6 to 3·0 with feathering wheels.

**Thickness of floats**, when of elm or other equally strong wood, is about $\frac{B}{12}$. Radial floats of common wood, $\frac{B}{8}$ is the thickness. Steel floats stiffened by curvature or angle bars are now often used instead of wood, as having less edge resistance and being cheaper. Their thickness should be 0·16 inch per foot of breadth + 0·16 inch.

*N.B.*—The edges, both of entry and emergence, should be chamfered at back, and the corners are better rounded.

Formerly wheels were made with two sets of rims, one pair being at the extremity of the arms outside the floats and the other pair just within the floats; in a modern wheel, with the big pitch of float possible with high rate of revolution and to reduce weight, the outer rims are omitted and the arms made as shown in Fig. 43.

**Immersion of floats** should not be less than one-eighth their breadth, and for general service should be one-half. If the voyage is so long that the fuel consumption materially affects the draught of water, or there is large variation due to other weight carried, the immersion at light draught should be not less than $0·1 \times B$, and when fully loaded not more than $0·7 \times B$. That is, for good running the variation in draught ought not to be more than 0·6B, which would mean 2·5 feet for a large ship and one foot for a small one.

**To design a feathering wheel** so that the floats shall enter edgeways when going at full speed, take P (Fig. 42, page 218), a point on the face of the float just entering the water, draw PA parallel to the water line, and cut off PA to represent the speed of the ship through the water, to some convenient scale; draw PB tangent to the circle through P, whose centre is the centre of the wheel, and cut off PB to represent the speed of the wheel on that circle; complete the parallelogram, and the resultant PR is the direction in which the float enters the water.

Produce RP, and draw parallel to it, at a distance from it equal to distance from gudgeon centre to face of float, a line cutting the circle of gudgeon centres in G.

Draw GH at right angles to PF, set off the breadth of the float, and mark off GL equal to the length of lever required.

Now draw another float whose face is vertical and immediately under the centre of the wheel, and the end of whose lever is M; and with centres M and L, and radius GC, draw two arcs of circles intersecting in E. Then E is the centre of the eccentric pin, and LE the length of the radius rods; and the ends of all the other float levers will lie in a circle struck from E as centre, with radius LE.

**Paddle-wheel frames.**—These must be strong enough, in every case, to transmit the power of the engines to the water, and must also be stiff enough to work without undue springing or vibration. As the power of the engines is approximately proportional to the cube of the diameter of the shaft, the strength of the wheel frames should be in the same proportion, and the section of the arm would be given by the equation,—

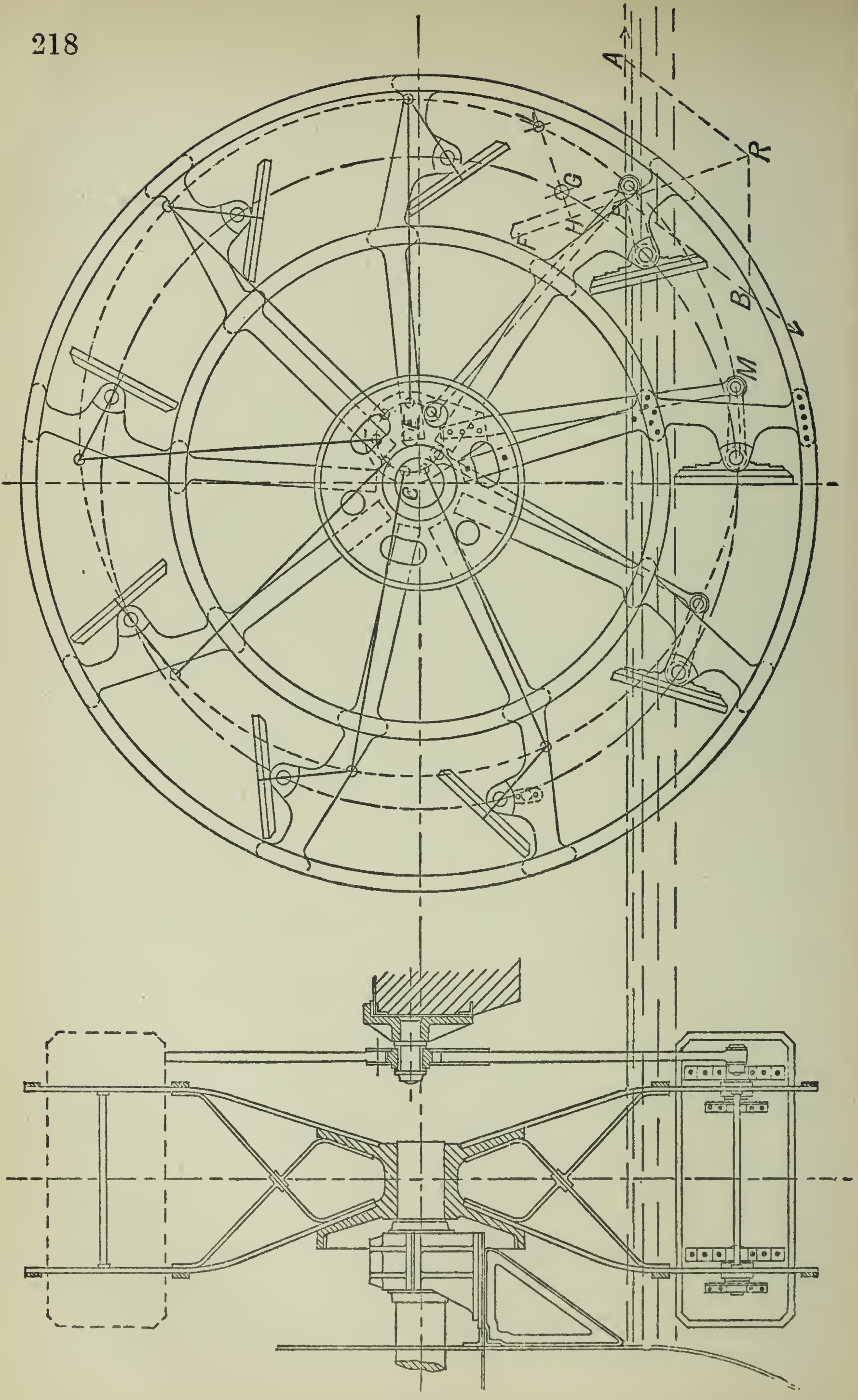

Fig. 42.

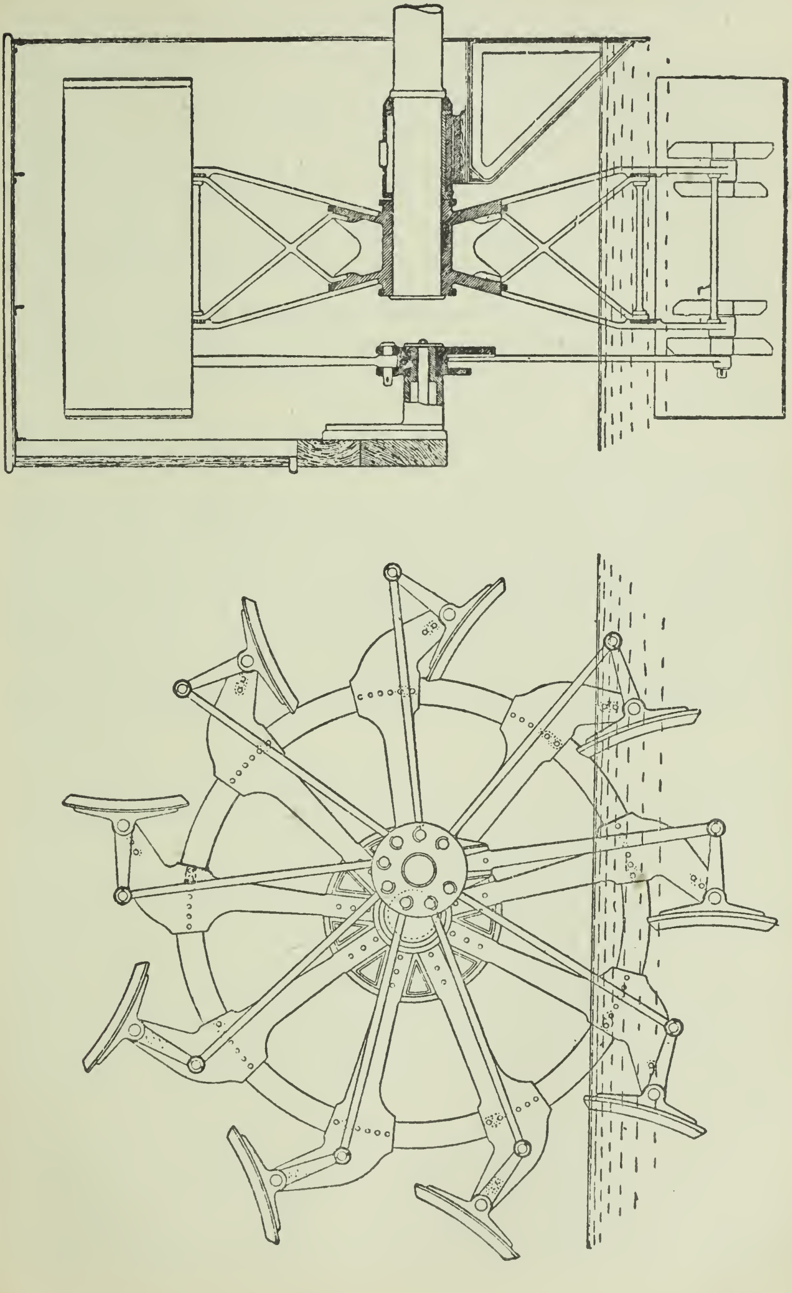

FIG. 43.—Paddle-Wheel with Steel Floats and no Outer Rim.

**Rule 202.** $$t \times b^2 = \frac{K}{n} \times d^3,$$

where $t$ and $b$ are respectively thickness and breadth of arm, $n$ total number of arms in one wheel, $d$ diameter of inner, or engine, journal of paddle-shaft, and K a co-efficient.

But when the diameter of shaft is small, this gives a section of arm which would not be stiff enough for practical purposes without more cross-bracing than is usual in small wheels; in practice, small wheels are commonly made without any diagonal bracing at all,—the arms being made proportionately stronger,—as experience has shown that it is not advisable to use sections below a certain degree of lightness.

The formula, therefore, requires the addition of a constant quantity, and, if written as follows, gives very satisfactory results:—

**Rule 202a.** $$t \times b^2 = \frac{K}{n} \times (d^3 + 600).$$

The value of K, for section of arm near boss, is 0·32,—and the other symbols are as defined above.

The formula may be used exactly as it stands for ordinary oscillating engines with sectional crank-shafts (Class 3, Table L.a), but the shafts of the other types are made larger, on account of the greater bending stresses to which they are subject, and what may be called the "equivalent cube of diameter" must be obtained before Rule 201 can be applied. The equivalent $d^3$ is found by taking the cube of the actual diameter of the journal, and, if for example the engine be of Class 2 (Table L.a), taking $\frac{50}{58}$ths of it.

Theoretically, the arms may be made lighter as they recede from the boss, but in practice it was usual to make them parallel, chiefly for convenience of manufacture, and to use rolled bar.

When there is no outer rim, or ring, the portion of each arm beyond the inner rim must be strong enough to carry its own load without any help from the other arms, and must be considered as a beam fixed at one end and loaded at the other. As there is a tendency to contrary flexure of the arm within the ring, the arm is generally made widest at the ring,—tapering gradually towards the boss, and quickly towards the gudgeon centre or end. The type of wheel is expensive, and the floats are deprived of such protection as is given by the outer rims, and are therefore so much the more liable to injury.

The ratio of breadth of arm to thickness is usually about 5 to 1. The arms must, of course, be locally widened out where the connection to the rim, or rims, is made. When there are two rims, from three to five bolts may be used at each joint, but with one rim only there should be six to ten bolts, according to size of wheel.

Nothing less than a ⅞-inch bolt should be used on ordinary wheels, and both heads and nuts should be square; also, all holes should be reamered out, and all bolts be driving fit.

**Rule 203.**

**The section of inner ring may be 0·7 × section of arm at boss.**

**Rule 203a.**

**The section of outer ,, 0·6 × ,, ,,**

**Rule 203b.**

**The section of only one ,, 1·0 × ,, ,,**

The proportions of breadth to thickness may be,—

**Rule 204.** Inner ring . . $\frac{b}{t}=5.$

**Rule 204a.** Outer ring . . $\frac{b}{t}=4.$

**Rule 204b.** Only ring . . $\frac{b}{t}=6{\cdot}5.$

In large wheels, the inner rims (or arms near inner rims) should be stayed back to the opposite sides of the boss; the section of stays may be,—

**Rule 205.**

**Section of diagonal stay (if rectangular) ·7 × section of inner rim.**

**Rule 205a.**

**Section of diagonal stay (if circular) ·5 × ,, ,,**

**Rule 206.**—Horizontal round ties, or distance rods, should also be fitted between each pair of gudgeon bearings, and as close as possible to them; they may have a diameter of 1·3 × thickness of arm.

It is a common practice to fix the centres of the gudgeons, on which the floats are hinged, slightly off the centre lines of the floats,—nearer the outer edges,—as the strains on the feathering gear are thereby reduced; the amount is to a great extent optional with the designer, but varies in practice from one-tenth to one-twentieth of the breadth of the float.

The whole of the gudgeons and pins should be cased with brass, and work in lignum-vitæ bushes; unless the vessel is to work in sandy or muddy water,—when iron pins working in white-metal bushes will give better results.

The outer bearing for the paddle-shaft may have a length of 1·5 to 2 × diameter of journal,—according to nature of service and consequent weight of wheel; it should be strongly made and firmly fixed, as, in addition to the weight of wheel, the whole of the thrust is taken by it. The magnitude and direction of the resultant force on the bearing can be easily calculated, but, as its direction is always below the horizontal, the caps do not need to be very substantial.

The lubricant was often mainly water, but oiling pipes and tallow boxes should be fitted, and water kept out.

A stuffing-box should be fitted round the shaft where it passes through the ship's skin.

## MARINE STEAM TURBINES.

**The turbine** is now employed on shipboard to drive the propellers in various ways. In warships it is practically the only motor now in use, and in the mercantile marine it continues to take the place of the reciprocating engine inasmuch as by means of gearing, single and double, the rate of revolution of the propeller can be reduced to permit of the employment of screws of large diameter and consequent high efficiency, especially in a seaway. It is a very cheap form of motor and simple in construction ; its weight is less than that of a reciprocator, and, while occupying no more floor space, it requires much less head room. As a rule it consumes less steam per power unit than the best of reciprocators, and it can make the fullest use of the high vacuum, so cheaply obtainable on shipboard. It can use steam both of high pressure and high temperature due to superheat, with some advantage, but these are not conditions necessary to it for economic or power results, inasmuch as the velocity obtainable with comparatively low initial pressures of steam is sufficient for good results.

The steam turbine is not such an elastic instrument for the marine engineer as the vertical reciprocator, inasmuch as by itself it is not reversible ; at low revolutions it is not so efficient and economic as at high, where the peripheral speed is in accordance with the velocity of flow of the steam ; the speed of revolution even at full speed is not generally so high as it should be for maximum efficiency, and the speed, such as it is, does not permit of an efficient screw propeller. That is, the turbine rate of revolution cannot be fixed arbitrarily by the designer as can be that of a reciprocator. If the rate of revolution is kept low, as it is on shipboard compared with what obtains on shore with turbines driving dynamos, there must be either an increase in diameter of the rotor so as to obtain the peripheral velocity ; or there must be a larger number of stages whereby the "drop" and consequent velocity of flow are reduced ; or there may be, as there generally was, on shipboard both these features, with the corresponding disadvantages of length of instrument and friction losses at the blades and channels.

The solution of these difficulties was attained by the introduction of gearing whereby the rate of revolution of turbine may be as high as desired or found practicable (now 3,000 to 4,000), while the screw may revolve at the rate for high efficiency under the service conditions. Double gearing is now in general use for large units, and likely to be so for even small ones, as the *spur* or main driving-wheel can thereby be of quite a moderate diameter.

**The reversing of the propellers** on a turbine-driven ship is accomplished by means of a separate turbine, either in a separate casing, etc., or housed in the rear of the ahead-going motor casing. The provision of power for this purpose is in Germany required to be 50 per cent. of the full power, with a consequent loss of efficiency of the ahead-going turbine of 3 to 5 per cent.; in England the astern-going power is not so great, being generally about 25 to 30 per cent., when the loss is about 1 per cent. There are now other methods of applying the turbine to propel ships, such as the electric and the hydraulic, whereby the reversal of the propeller is attained without a separate motor. The efficiency of such installations varies from 0·865 of small powers (150 S.H.P.) to 0·900 of the large power in express steamers. So that the losses due to their mechanism are more rather than less those of the astern-going turbine.

The turbines in use in warships and express steamers are the *Parsons*, the *Curtis*, the *Zoelly*, and the *Rateau*. Their efficiency, as judged by steam consumption per power unit on shore, is as high as 70 per cent., but generally it is somewhat less on shipboard. Their consumption of steam at full speed per S.H.P. hour, when of large size, is as low as 12 lbs. under conditions with which the theoretical maximum output of a lb. of steam is 7·64 H.P. (*vide* Table V.).

$$\text{That is, Efficiency} = \frac{60}{12} \div 7{\cdot}64 = 0{\cdot}654.$$

The arrangement of turbines in ships is now as follows:—

(1) A complete head-going machine with all its stages coupled direct to a propeller shaft, and having an astern-going member in the rear of it, both contained in a common casing.

(2) The turbine is divided into two parts, the high-pressure or boiler end is in a separate casing and coupled to its own screw shaft direct; the low-pressure or condenser half has its own casing and is coupled to its own screw shaft, while within the casing is the astern-going turbine. In some cases there is an astern-going turbine in an independent casing, etc., in rear of the high-pressure half.

(3) There is a high-pressure portion in its own independent casing coupled to its own propeller shaft, while there are *two* low-pressure portions, each having its own casing with a reversing turbine, and each coupled to its own propeller shaft.

(4) Each propeller shaft is fitted with a pair of helical steel wheels, side by side, right- and left-handed, and on each side of this

composite spur wheel is a pinion and shaft, one driven by the high-pressure half and the other by the low-pressure half of a compound turbine, as in No. 2.

(5) Each propeller shaft is fitted, etc., as in No. 4, but the rotor shaft is not coupled direct to the spindle which has the pair of pinions. but has interposed a spindle with a pair of pinions which gear into a spur wheel on that spindle.

(6) One or more complete turbines are coupled either direct or by means of gearing to a dynamo, the electricity developed by which is delivered to a motor or motors fitted to the screw shafts direct.

(7) One or more complete turbines are caused to drive some form of pump which delivers its water to a hydraulic motor connected direct to the screw shafts.

In every case the low-pressure member exhausts to an independent condenser, with air, circulating pumps, etc. Direct drive is now a thing of the past with turbines; single gearing involves very large spur wheels to reduce the rate of revolution of the screw to practical limits, and when that rate is that best for large ships, double gearing is a necessity and is now the general practice. The older ships with direct drives followed Nos. 1, 2 and 3 methods.

No. 1 method was the usual one with Destroyers, Scouts, and small ships of very high speed. No. 6 is much favoured in the U.S.A. Navy, while No. 7 was adopted by the Germans in 1913.

The low-pressure portion of the turbine is by far the most effective, and its efficiency is considerably higher than that of a reciprocating engine; on the other hand, the high-pressure portion is not so efficient as the reciprocator. The turbine can also develop a considerable amount of work from the steam as it exhausts from the low-pressure cylinder of a triple compound engine when working at full power, as in it steam can expand to within half a pound of that in the condenser, and that may be with a turbine as low as half a pound (or 29 ins. vacuum) with advantage. Under these circumstances a low-pressure turbine may be attached to a triple compound engine with a considerable increase in power occurring, or the same aggregate power as developed by the triple compound engine may be obtained from the combination with less steam. The gain is usually 15 to 20 per cent. in shore installations. On shipboard this arrangement is easily effected with three screws; there may be, as in the R.M.S. *Olympic*, two reciprocators driving the wing screws, with a turbine taking the exhaust steam from the two reciprocators and driving the centre screw. Or there may be one centre reciprocator and two wing turbines; this is, of course, not so suitable an arrangement except as a means of further compounding a single-screw steamer. The economic result of such a combination can be very good, as the comparative inefficiency of the high pressure end of the turbine is replaced by the more efficient triple reciprocator, and the comparatively inefficient low-pressure end of what would be a quadruple reciprocator is replaced by the highly efficient low-pressure turbine.

The arrangement of two reciprocators with twin screws exhausting to a low-pressure turbine driving the centre screw is in every way a convenient one for both merchant ships and war vessels, for the reciprocators are reversible, and so all manœuvring can be done by them with the turbine out of action, and likewise cruising can be done economically. When full speed is desired the exhausts can be easily and quickly switched to the turbine. As a matter of fact, with the reciprocators out of action quite a decent "slow speed" can be maintained by turning the exhaust from all the auxiliaries to the turbine; if supplemented with a modicum of steam direct from the boilers, a better speed can be maintained when required.

This, which may be called the eighth method of applying the turbine system to ship propulsion, is now a popular one, and being used more especially for passenger steamers of moderately high speed working where fuel is dear. The s.s. *Otaki*, of 6857 I.H.P., built by Denny Bros., Dumbarton, having the combination, showed a gain on fuel consumption of 17 per cent. over a sister ship with triple compound reciprocators only.

The steam consumption at full power was 11·95 lbs. per I.H.P. hour. The efficiency under the circumstances was $\frac{60}{11{\cdot}95} \div 7{\cdot}9 = 0{\cdot}635$.

The steam consumption on the main turbine engines of R.M.S. *Lusitania* was 12·77 lbs. per S.H.P. hour. The efficiency in her case was

$$\frac{60}{12{\cdot}77} \div 7{\cdot}65 = 0{\cdot}614.$$

**Turbines geared to the Propeller Shafts** is the method now in favour as the means whereby the turbine may be used for cargo ships, and the trials made by Sir C. Parsons with the s.s. *Vespasian* amply demonstrated the practicability of it for such a purpose. Since then further experiments with cargo steamers have confirmed the opinions formed by Sir C. Parsons and his friends. But the most interesting development had been the fitting of a cross-channel express steamer with twin-screws driven by turbines geared, and trying her against similar three-screw boats, direct-driven turbine boats, on the same station. The following were the results: the higher efficiency of the geared installation is no doubt due to the fact that by such means the turbines can be run at rates of revolution approaching those on electric generating plants on shore, while the screws can be of such a size and rate of revolution as to give high efficiency of propeller. The efficiency of the *Normania* machinery as measured by steam consumption is $\frac{60}{12} \div 7{\cdot}60$ or 0·658, as against $\frac{60}{15{\cdot}1} \div 7{\cdot}60$ or 0·526 of the *Sarnia*.

Professor Biles states that only 1½ per cent. of the power was lost in the gearing. That is, its efficiency was 0·985. This is, of course, very satisfactory, and so long as no abrasion takes place it will continue to be so. The wheel and pinions are machine cut, with great accuracy, and the latter run in an oil bath.

### Table LXX.—Performances of Channel Steamers with Reciprocators and Turbines.

| Particulars of Ship and Machinery. | Twin S.S. P.A. Recipros. | Twin S.S. Chfd. Recipros. | Three S.S. *Sarnia*, Direct Turbines. | Twin S.S. *Normania*, Geared Turbs. |
|---|---|---|---|---|
| Length between perpendiculars . . . . . ft. | 290 | 300 | 284 | 290 |
| Breadth . . . . . ,, | 38·0 | 34·5 | 39 | 36·0 |
| Draught of water . . . ,, | 11·93 | 13·8 | 12·0 | 12·0 |
| Displacement . . . tons | 2,100 | 2,200 | 1,990 | 1,875 |
| Block co-efficient of fineness . | 0·581 | 0·559 | 0·524 | 0·524 |
| Wetted skin . . . sq. ft. | 12,400 | 13,000 | 11,900 | 11,700 |
| Engines . . . . . . | 4 Crank Triple | 3 Crank Triple | Direct Turbines | Geared Turbs. |
| Total heating surface . sq. ft. | 11,450 | 11,450 | 12,985 | 10,221 |
| Boiler pressure . . . lbs. | 180 | 160 | 160 | 160 |
| Propellers, diameter of . ft. | 10·5 | 12·0 | 5·50 | 8·0 |
| ,, pitch . . ft. | 14·5 | 16·0 | 4·52 | 7·12 |
| Revolutions of engines . . | 164 | 130·4 | 500 | 1984 & 1380 |
| ,, ,, screws . . | 164 | 130·4 | 500 | 310 |
| Horse-power developed . . | 5,892 | 4,398 | 6,670 | 5,000 |
| Speed, knots . . . . . | 19·12 | 18·20 | 20·0 | 19·7 |
| $\text{Displacement}^{2/3} \times \text{Speed}^3 \div$ I.H.P. | 194 | 232 | 190 | 232 |
| I.H.P. per 100 ft. of wetted skin | 47·5 | 33·8 | 56·1 | 42·8 |
| ,, ,, ,, reduced to 10 knots . . . . . . | 6·79 | 5·60 | 7·01 | 5·61 |
| Steam consumed per H.P. hour . . . . lbs. | 16·75 | 16·30 | 15·1 | 12·0 |
| Coal consumed per H.P. hour ,, | ... | ... | 1·71 | 1·43 |

**The rate of revolution** of a marine turbine can, within limits, be determined arbitrarily, for there is no question of periodicity to complicate matters. There is, however, propeller efficiency to consider, both theoretically and practically, and it is here that the designer finds his limits. He may choose a screw of small diameter with very large surface ratio and quite a good pitch ratio, or he may, for practical reasons, adopt one of larger diameter, when the pitch ratio will be smaller and the surface, as well as surface ratio, smaller; both suitable to the revolutions fixed on for the turbines. As a rule, the screw of a turbine ship is a compromise between these conditions. If a diameter, such as one would choose for a reciprocator, were used, the pitch ratio would be smaller than consistent with efficiency, and the surface small too—so small that the actual slip would be *less* than that given by revs. × pitch,—the apparent slip would be as high as that observable in a twin-screw ship driven by one screw only.

The actual slip ratio may be, of course, as high as desired by making the diameter small, so that the column of water is of small section—in practice, however, such screws would be inefficient in sea service and be bad for manœuvring.

The following rule may be observed for arriving at a suitable diameter for actual sea service in direct-drive turbine ships.

**Rule 208.** **Diameter of screw** $=\sqrt{\dfrac{\text{S.H.P.}}{x-x^2}\times\left(\dfrac{\text{C}}{\text{P}\times\text{R}}\right)^3}$.

Here P is the pitch in feet; R the revolutions per minute; $x$ is the *real* slip ratio and may be taken as $=0{\cdot}2\ \text{P}_c+f$, $\text{P}_c$ being the prismatic co-efficient of the ship's immersed body, and $f$ is the apparent slip. The value of C is 450. The centre screw, especially of small ships, may be of larger diameter based on $x=0{\cdot}18\ \text{P}_c+f$.

The following, however, is a simpler rule and may be used with confidence:—

**Rule 208a.** **Diameter of screw** $=z\times\text{P}_c\sqrt[3]{\dfrac{\text{S.H.P.}}{\text{R}}}$.

Generally $z=5{\cdot}75$ for twin and $6{\cdot}55$ for centre screws (when three).

For large ocean-going steamers, $z=6{\cdot}5$ for all three or four screws.

For T.B. destroyers, $z=5{\cdot}4$.

**The slip ratio** in common practice is 13 to 15 per cent. with large ocean-going expresses, and 20 to 22·5 per cent. with smaller cross-channel ships.

**The pitch ratio** with turbines direct driven is very seldom over 1·0, and generally under; 0·9 to 0·95 is considered good practice.

**The actual rate of revolution** will be found from these fundamental conditions, and in practice it varied from 185 of the Atlantic expresses to 750 of the cross-channel ones. The *Lusitania* and *Mauretania* did over 190 revolutions at full power with their original screws, and 180 to 185 with new screws of larger pitch ratio.

In the Mercantile Marine, very high-power ocean expresses 185 revolutions, smaller ones 200. Large cross-channel expresses 450 to 500 revolutions, smaller ones 600 to 750, and small excursion and river steamers 750 to 1000.

The tendency latterly had been to reduce the number of revolutions and increase the diameter of screw.

With geared installations it is possible to have a rate of propeller revolution approximating to that which obtained with reciprocators. Our largest and most powerful warships, with an aggregate power over 100,000 S.H.P., have a rate of about 340 r.p.m. with a turbine velocity of about 3000 revolutions. The large express steamers are run at a rate of about 130 r.p.m., while with the ordinary cargo or cargo-passenger steamer it is as low as 80 to 100 r.p.m. On the cross channel or small express steamer the general practice is 250 to 300 r.p.m.

**The diameter of the shafting** is, of course, small for the total power, but is based on the power per revolution, which is small. The rotor shaft has to be designed to suit the weight of the rotor and to give the necessary bearing surface. The low-pressure rotor of the *Olympic* weighs complete 132 tons, and has journals 33 inches in diameter; its diameter is 195 inches, and length over all 47·6 feet. That of the *Lusitania* was 184 inches in diameter over all, and its weight 120 tons; it was carried on journals 33⅛ inches diameter and 56½ inches effective length, so that the pressure per square inch was 72 lbs. only, while the speed of rubbing surface was 1650 feet per minute. The pressure on the journals of the high-pressure rotor was 80 lbs., but the rubbing speed only 1350 feet per minute.

**Rule 209. Working pressure per square inch of journal** $= 3000 \div \sqrt{S + 100}$ may be taken as a guide in determining the length of journals. Their diameter must be calculated from considerations of torque transmitted, bending moment due to weight, and, to some extent, to inertia, and always with regard to stiffness and the critical speed.

The rules for shafting of turbines are as follows:—

**Rule 210. Diameter of Intermediate Shafts** $= \sqrt[3]{\frac{\text{S.H.P.}}{\text{R}}} \times \text{F}$.

| | | |
|---|---|---|
| The value of F by | Board of Trade is . . | 64 |
| ,, ,, | Lloyd's Register is . . | 70 |
| ,, ,, | British Corporation is . | 65 |
| ,, ,, | Bureau Veritas is . . | 70 |
| ,, ,, | B.M.E.D. & C. Committee is | 64 for ocean going |
| ,, ,, | ,, ,, | 58 smooth water |
| ,, ,, | Admiralty is . . . | 32 to 60 |

The diameter of thrust-shaft near collars and of the inner end of the tube shaft must be 5 per cent. larger in diameter. The tail-end shaft has added to the diameter of the intermediate shafts.

**Rule 211. Diameter Tail Shaft** = diameter intermediate shaft $+ x \times$ diameter of screw. Value of $x$ is 0·006 for Bureau Veritas, and 0·008 for B.M.E.D. & C. Committee.

The Board of Trade multiplies the diameter of the intermediate shaft by 1·112, the British Corporation by 1·10. Lloyd's rule is diameter of intermediate shaft $\times \left(0{\cdot}63 + \frac{0{\cdot}03 \times P}{d}\right)$, with a limitation of 1·07 × diameter intermediate shaft.

Q = 1860 ÷ tensile ultimate strength of metal in tons per square inch, and make the above additions for propeller shafts.

**The rotor drums** are generally of forged steel, but often latterly steel castings. They must be most carefully and accurately balanced, especially when the latter is used. The arms or discs which connect the drums to the axle must be stiff enough to prevent springing or distortion, and consequently, then strong enough to transmit the torque. The rate of revolution of marine turbines is so low compared with those obtaining on shore work that centrifugal forces and the stresses produced are moderate, but still large enough to demand great care in design and construction. With geared turbines these stresses are, of course, much higher.

**The blades of both rotors and stators** are usually of bronze, one of the strong zinc-bronzes capable of being rolled, drawn out, or extruded, of the required section, being preferred. These alloys, however, while being strong against tensile forces and soft enough to stand manipulation, are weak against shearing forces ; phosphor-bronze is therefore preferred by some engineers, as this can be drawn into strips, while possessing a hardness and high elastic limit superior to the zinc-bronzes. It will also stand better the eroding action of steam at high velocity. Steel blades are also fitted, especially where sufficient section cannot be got in the ordinary way to permit of the bronze. Steel alloys—especially the nickel ones—have been used. Corrosion, however, is not altogether unknown—due often to leaky condenser or other similar defects admitting sea-water to the boilers. Too much care cannot be devoted to the workmanship when blading the rotors, and the same may be said of the stators, but the latter are not subject to such severe trials as the former. Each builder of turbines has his own design of blade, choice of material, and method of fixing. The strip system is simple, inexpensive, and quite effective when properly done ; at high rates of revolution, with corresponding high centrifugal action on them, they are not always so trustworthy ; in the past, with direct coupled turbines on shipboard, this has not had to be dwelt on, now, however, with geared ones at 2000 revolutions, the same attention must be given to it as obtains on shore. Since high vacuum steadily and cheaply maintained is essential to the economic working, it is absolutely necessary there should be no air leaks anywhere, and the low-pressure turbine journals should be provided with means to prevent air passing through them when they are leaky ; water or steam passing are of no consequence, so provision may be made for one of them to seal the journal.

**The design of a turbine** is really a simpler problem than that of a compound reciprocator ; it is, however, started in quite a different way, except that in both cases the intended rate of revolution is assumed. With the turbine it is usual to begin with a rate of steam consumption per unit per hour ; with an initial pressure of 175 lbs. absolute, 15 lbs. is a covering allowance with a vacuum of 28 inches. At higher initial pressures the available work is, of course, greater ; but with them the turbine is not so efficient.

Quantity of steam per second per 1000 S.H.P. $= \dfrac{1000 \times 15}{60 \times 60}$, or 4·17 lbs.

Of this, from 3 to 5 per cent. will disappear by leakage, so that the nett amount through first guide blades will be only about 4·0 lbs., or 10·28 cubic feet. If the flow there is 250 feet per second, and the rate of revolution is 480 per minute or 8 per second,

Area through guide blades $= \dfrac{10{\cdot}28 \times 144}{250} = 5{\cdot}92$ square inches, and so on throughout the various stages during which the steam is expanding, acquiring velocity, and giving out its kinetic energy to the rotor blades.

The loss of area through the annulus, in which the blades act due to the blade section, is so considerable that it may be taken that:—

The gross area $= 5{\cdot}92 \div 0{\cdot}32$ per 1000 S.H.P., or 19 nearly.

**Rule 213.**

$$\textbf{Mean diameter of rotor in inches} = \frac{\text{Velocity of steam} \times 12 \times 60 \times f}{\text{Revs.} \times \pi}.$$

Factor $f$ is the ratio of the velocity of the blades to that of the steam, and is usually 0·4.

In this particular case,

$$\text{Mean diameter} = \frac{250 \times 12 \times 60}{480 \times 3{\cdot}1416} \times 0{\cdot}4 = 47{\cdot}7 \text{ inches.}$$

Height of blade $= 19 \div \pi \times 47{\cdot}7$, or 0·127 in. per 1000 S.H.P. In practice, such short blades would not be fitted; instead, a considerable portion of the first ring on the stator would be blocked, leaving ports in which the blades would act.

**Rule 214. The size of exhaust passage** is, of course, very great, for, if it is assumed that the flow to condenser is at a pressure of 1½ lbs., the volume will be about 900 cubic feet, which, with a rate of flow of 160 feet per second, the area of section will be 810 square inches per 1000 S.H.P.

**Rule 215.** The length of blades should never exceed 15 per cent. of the diameter of rotor; from 8 to 10 per cent. gives the best results. Sometimes they are only about 1 per cent., in which case the leakage is bound to be high, inasmuch as the clearance space is high compared with the blade annulus.

The losses of a marine turbine have been summarised by Mr S. J. Reed as follows:—

$$\left.\begin{array}{l}\text{Leakage over dummies}\\ \text{Friction of bearings, glands, \&c.}\\ \text{Radiation}\\ \text{Loss by carry over of kinetic energy from last}\\ \quad\text{row of blades of low-pressure turbine}\end{array}\right\}\text{8 to 12 per cent.}$$

| | |
|---|---|
| Friction of steam in blade channels<br>Leakage over tops<br>Steam shock and eddying<br>Loss of kinetic energy due to splitting action of blades<br>Loss of carry over from last row blades of high pressure | 35 to 45 per cent. H.P.<br>25 to 35 per cent. L.P. |

**Testing of turbine** cases is made by water pressure after being machined, but before blading, &c.

**Rule 216.** *The Admiralty require* the test pressures to be as follows:—

(1) High-Pressure Turbine, ahead and astern.

| | |
|---|---|
| Steam end and inlet cover | 255 lbs. |
| The remainder | 200 ,, |
| After completion | 170 ,, |

(2) Cruising Turbine.

| | |
|---|---|
| Steam end and inlet cover | 255 lbs. |
| The remainder | 200 ,, |
| After completion | 170 ,, |

(3) Low-Pressure Turbine.

| | |
|---|---|
| Astern cylinders and inlet cover | 50 lbs. |
| Ahead cylinders and inlet cover | 30 lbs. |
| Exhaust chambers and passages | 30 ,, |
| After completion | 30 ,, |

| | |
|---|---|
| (4) Receiver pipe and valve between cruising and high-pressure ahead turbine | 255 lbs. |
| (5) Steam chambers of condenser and eduction pipes | 30 lbs. |

**Rule 217.** *The Board of Trade require* high-pressure turbine cases to be tested to $1{\cdot}33 \times$ working pressure. Low-pressure cases, condenser end, 30 lbs. The astern going cases to working pressure.

**Rule 218.** *The Italian Government demands* are somewhat similar, viz.:—

Casing of high-pressure turbine to $1{\cdot}33 \times$ working pressure; astern going to working pressure; low-pressure turbine, admission end, $0{\cdot}33 \times$ working pressure, and exhaust end 28·5 lbs. per square inch. Condensers for turbines 28·5 lbs.

The weights of turbine installations are less than that of reciprocators.

**Rule 219.** Total weight of turbine machinery (Naval) with water-tube boilers

$$= \frac{\text{S.H.P.} \times 16{\cdot}85}{\text{Revs. per minute} - 100} \text{ tons.}$$

Mercantile installations on Board of Trade rules, with cylindrical boilers and full complement of auxiliaries, as in first-class steamers:

**Rule 220. Total weight machinery** $= \dfrac{\text{S.H.P.} \times 44}{\text{Revs. per minute} + 50}$ tons.

(i) *Example.*—Cross-channel steamer, 8000 S.H.P., revs. 500.

$$\text{Total weight} = \frac{8000 \times 44}{550} = 640 \text{ tons or } 179{\cdot}2 \text{ lbs. per S.H.P.}$$

(ii) *Example.*—Naval scout of 28,000 S.H.P. at 750 revs.

$$\text{Total weight} = \frac{28{,}000 \times 16{\cdot}85}{750 - 100} = 726 \text{ tons, or } 58{\cdot}1 \text{ lbs. per S.H.P.}$$

**The steam consumption of a turbine** ought, according to Professor Rateau, to be nearly as given by his formula.

**Rule 221.**

$$\textbf{Steam consumption} = 2{\cdot}13 + \frac{16{\cdot}2 - 2{\cdot}05 \log_e P}{\log_e P - \log_e p} \text{ lbs. per hour}$$

P being the initial and $p$ the final pressures.

**The power developed** by and transmitted from a turbine is found by ascertaining the torque or twisting moment from observations of the twist or angular displacement of a definite length of shaft. Then for shaft diameter $d$:—

If $T_1$ is the torque in inch-lbs., R the revolutions per minute, $\theta°$ the angular displacement in degrees,

$$\text{S.H.P.} = \frac{T_1}{12} \times \frac{2\pi R}{33{,}000} = \frac{T_1 \times R}{63{,}000},$$

$$\text{or} \qquad T_1 = \frac{\text{S.H.P.}}{R} \times 63{,}000.$$

$$\text{Now} \qquad T_1 = \frac{\theta° \times 20{,}120\, d^4}{L} \text{ for solid shafts, and}$$

$$T_1 = \theta° \times 21{,}147 \left(\frac{d^4 - d_1^4}{L}\right) \text{ for hollow with bore } d_1.$$

It follows on combining these equations that

**Rule 222.** $\quad (1)\ \text{S.H.P.} = \frac{R}{L} \times \frac{d^4}{3{\cdot}13}$ for solid.

**Rule 222a.** $\quad (2)\ \text{S.H.P.} = \frac{R}{L} \times \frac{d^4 - d_1^4}{2{\cdot}98}$ for hollow shafts.

L is the length of shaft under observation, generally taken as 40 inches to suit the usual torsionmeter.

The formula in every day use is as follows :—

**Rule 223.** $\quad \text{S.H.P.} = \frac{\theta° d^4 \times R}{Q \times L}$ for solid, and $\frac{\theta° \, (d^4 - d_1^4) \times R}{Q \times L}$ for hollow shafts, where Q is taken as 3·27, the modulus assumed being 11,250,000, which is for the ordinary mild-steel shafts of commerce. The Admiralty require a somewhat stronger steel (32 tons ult-tensile), and always have the actual shaft tested mechanically with its own torsionmeter in the shops before fitting into the ship, to ascertain the value of $\theta°$ for a series of twisting moments. From these observations a curve for the value of $T_1$ can be made and the value of the factor determined for each shaft and ship.

The **torsionmeters** in use are either wholly mechanical, like the Föttinger, which can, and does actually register the torque by a continuous line on a sheet of paper like an indicator diagram; or they may be partly mechanical, the registering being made by a beam of light from a mirror actuated by the mechanism thrown on a scale, which can be read off while the shaft is running. Hopkinson and Thring's instrument is of the latter kind, and gives good and reliable results. Then there is the Bevis-Gibson apparatus, where a beam of light is transmitted through slots in discs, two on the shaft and one fixed. These are in line when the shaft is at rest or running without load, and can be brought into line when running under load by the angular displacement of the fixed disc. The amount of displacement necessary is the measure of the twist. There is a fourth kind, in which synchronism is indicated by sound, and the production of this condition when running under load is effected in such a way that the compensation indicates the twist. This latter is the Denny-Johnson.

## Trials of Turbine-driven Ships.

The following results of trials with ships fitted with turbines, or reciprocator-turbine arrangements, are interesting and instructive.

(1) R.M.S. *Lusitania*, a four-screw Atlantic express steamer, 760 feet long × 87·5 feet beam, and on a draught of water of 31·6 feet (mean) displaced 35,600 tons (*v.* Table X.).

### Steam and Fuel Consumptions of R.M.S. "Lusitania," with Turbines running on Sea-service Conditions at Various Speeds.

| Speed of Ship in Knots per Hour. | 15·77 | 18·00 | 21·00 | 23·00 | 25·40 |
|---|---|---|---|---|---|
| Shaft horse-power . . . . | 13,400 | 20,500 | 33,000 | 48,000 | 68,850 |
| Consumption of steam per hour, total . . . . . . lbs. | 284,500 | 253,600 | 493,300 | 668,300 | 879,500 |
| Consumption of steam auxiliaries lbs. | 71,000 | 76,400 | 85,700 | 96,700 | 116,500 |
| Consumption of auxiliaries per cent. of total . . . . . | 25·0 | 21·6 | 17·4 | 14·51 | 13·2 |
| Steam consumption per S.H.P. hour, turbines . . . lbs. | 21·23 | 17·24 | 14·91 | 13·92 | 12·77 |
| Steam consumption per S.H.P. hour, auxiliaries . . . lbs. | 5·30 | 3·72 | 2·60 | 2·01 | 1·69 |
| Steam consumption per S.H.P. hour, total . . . . lbs. | 26·53 | 20·96 | 17·51 | 15·93 | 14·46 |
| Coal consumed per S.H.P. per hour, total . . . . lbs. | 2·52 | 2·01 | 1·68 | 1·56 | 1·43 |
| Total consumption of fuel on voyage, 3100 miles . tons. | 2980 | 3190 | 3670 | 4520 | 5390 |
| Time on voyage . . hours | 203 | 178 | 152 | 139 | 126 |

(2) S.S. *Otaki*, a three-screw passenger and cargo steamer for New Zealand service. She is 464·5 ft. × 60 ft. × 34 ft., and on a mean draught of 27·5 feet displaces 9900 tons. Her trials, however, were carried out on 20 feet mean draught. She was fitted with twin screws, each driven by triple reciprocators having cylinders 24·5 inches, 39 inches, and 58 inches by 39-inch stroke, exhausting into a low-pressure turbine, 90 inches diameter, driving a central screw. The following shows the results of progressive trials and a comparison with the performance of a sister ship, s.s. *Orari*, driven by twin screws, each having cylinders 24·5 inches, 41·3 inches, and 69 inches by 48-inch stroke.

## Measured Mile Trials of S.S. "Otaki," with Mixed Machinery, 31st October 1908, on 20 feet Mean Draught.

| | Mean of A Runs. | Mean of B Runs. | Mean of C Runs. | Mean of D Runs. |
|---|---|---|---|---|
| Total horse-power, being I.H.P. (reciprocators) + S.H.P. (turbine) . . . . | 6857 | 5348 | 4704 | 3282 |
| Mean speed . . . . . . knots | 15·02 | 14·28 | 13·83 | 12·52 |
| Revolutions, reciprocators . . . . | 103·5 | 97·9 | 93·5 | 83·4 |
| ,, turbine . . . . . . | 224·5 | 209·7 | 197·2 | 172·1 |
| Total water consumption per hour . lbs. | 82,000 | 67,300 | 60,200 | 44,600 |
| Total water consumption per hour, per H.P. . . . . . . . lbs. | 11·95 | 12·6 | 12·8 | 13·6 |
| Mean absolute pressure at H.P. cylinder lbs. | 193 | 178 | 166 | 135 |
| ,, ,, turbine inlet lbs. | 9·5 | 7·62 | 6·76 | 5·0 |
| Vacuum at exhaust end of turbine . . | 28·1 | 28·2 | 28·4 | 28·5 |
| ,, on condenser gauge . . . | 28·2 | 28·4 | 28·3 | 28·5 |
| Temperature of sea-water . . . F.° | 56 | 56 | 56 | 56 |
| ,, circulating discharge . . | 70 | 67 | 70 | 70 |
| ,, hot well . . . . . | 72 | 70 | 73 | 74 |
| Steam consumption based on the I.H.P. of s.s. *Orari* and measured by tanks . . | 13·66 | 13·7 | 13·8 | 13·07 |
| As measured by pumps per I.H.P. per hour | 14·12 | 14·1 | 14·3 | 15·2 |

| Name of Ship. | E.H.P. | I.H.P. | Propulsive Co-efficient. | Water Consumption per Hour. | | |
|---|---|---|---|---|---|---|
| | | | | Total. | Per E.H.P. | Per I.H.P. |
| | | | Per cent. | | | |
| 3-screw s.s. *Otaki* (turbo-reciprocators) . . | 3350 | 5880 | 57 | 73,300 | 21·9 | 13·7 |
| 2-screw s.s. *Orari* (reciprocators) . . . | 3210 | 5360 | 60 | 88,300 | 27·5 | 16·5 |
| Gain per cent. in *Otaki*. | .. | .. | .. | 17 | 20 | 17 |

(3) The two ships compared here are channel steamers of the L. & S.-W. Ry. Co., both driven by turbines (*v.* Table LXX.).

(*a*) S.S. *Sarnia*, a three-screw steamer, is 284 ft. × 39 ft. × 23·8 ft. moulded, and on a mean draught of water of 12 feet displaces 1990 tons. She has three screws, driven direct by the usual installation of high-pressure and low-pressure turbines running at about 500 revolutions per minute. Her speed, with 6670 S.H.P. developed, is 20 knots, giving an Admiralty co-efficient of 190 only.

(*b*) The S.S. *Normania*, a twin-screw ship, is 290 ft. × 36 ft. × 23·5 ft. moulded and a displacement of 1900 tons on 12 feet draught. She has two screws, each driven by two turbines geared by pinion and spur wheels to the propeller shafts, and maintained a speed of 19·7 knots with 5000 S.H.P., showing a co-efficient of 232·4, or 22·5 per cent. higher than the *Sarnia*. The fuel consumption per voyage of the *Normania* is, however, as much as 40 per cent. less than that on the *Sarnia*. This vast difference is due to three causes : (i) the superior efficiency of the turbines, due to the 1984 revolutions per minute of the high pressure and 1380 of the low pressure ; (ii) the superior efficiency of screws, 8 feet diameter, running at 300 revolutions, over those of only 5·5 feet diameter at 500 revolutions ; and (iii) to the better proportion and form of hull with the twin screws. The total steam consumption of the *Sarnia* was as much as 17·1 lbs. per S.H.P. hour, whereas the *Normania* used only 14·3 lbs., or 16·4 per cent. less.

(4) The trials of H.M.S. Cruisers *Amethyst*, driven by turbines and three screws, and the *Topaze*, by twin screws and triple reciprocators, are interesting, but hardly fair to the latter, as her engines had cylinders, etc., for great power without regard to economy of consumption ; so that it was only at reduced speed that economic conditions existed. As showing the variation of the two systems the following is perhaps more interesting.

## Water Consumption per I.H.P. per Hour of H.M.S. "Amethyst" and "Topaze" on Progressive Trials.

| Speed in Knots. | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| H.M.S. *Amethyst* (turbines) . | 29·3 | 26·0 | 23·7 | 22·0 | 20·4 | 19·0 | 17·9 | 16·8 | 15·9 | 15·2 | 14·7 | 14·3 | 14·0 |
| H.M.S. *Topaze* (reciprocators) . | 23·8 | 22·0 | 20·6 | 19·6 | 18·8 | 18·4 | 18·3 | 18·4 | 18·7 | 19·2 | 19·8 | 20·5 | 21·4 |

(5) The series of trials of the Cruisers, "City" Class, is instructive. These ships are alike and 430 ft. × 47 ft. × 26·75 ft., having a displacement

of 4800 tons and 15·25 feet mean draught. The *Bristol* has Curtis turbines and twin screws, the others the ordinary Parsons turbines with four screws. Each ship has twelve Yarrow water-tube boilers. The following are the results of the various trials:—

### Trials of the Quadruple Screw Cruisers, "City" Class.

| | H.M.S. *Glasgow.* | H.M.S. *Liverpool.* | H.M.S. *Newcastle.* | H.M.S. *Gloucester.* | H.M.S. *Bristol.** |
|---|---|---|---|---|---|
| A. 22 hours at 66% full power. | | | | | |
| S.H. Power . . . | 14,055 | 14,102 | 14.051 | 13,968 | 14,300 |
| Revolutions per minute . | 430 | 426·5 | 420·5 | .. | .. |
| Mean speed, knots . . | 23·70 | 23·88 | 23·34 | 23·45 | 24·06 |
| Coal per S.H.P. hour . | 1·61 lbs. | 1·57 lbs. | 1·65 lbs. | 1·59 lbs. | .. |
| B. 8 hours at 80% power. | | | | | |
| S.H. Power . . . | 18,220 | 18,824 | 19,116 | 18,983 | 19,130 |
| Revolutions per minute . | 471 | 464 | 461·6 | .. | .. |
| Mean speed, knots . . | 24·90 | 25·10 | 24·84 | 25·08 | 25·17 |
| Coal per S.H.P. hour . | 1·57 lbs. | 1·59 lbs. | 1·65 lbs. | 1·48 lbs. | .. |
| C. 8 hours at full power. | | | | | |
| S.H. Power . . . | 22,500 | 24,718 | 25,417 | 24,340 | 24,230 |
| Revolutions per minute . | 499 | 512·7 | 518·2 | .. | .. |
| Mean speed, knots . . | 25·80 | 26·17 | 26·27 | 26·31 | 26·84 |
| Fuel per S.H.P. hour . | 1·63 coal | 1·65 coal | 1·036 coal<br>·171 oil | 1·14 coal<br>·42 oil | |
| | | | 1·207 | 1·56 | .. |
| $D.^{2/3}\times S.^{3}\div$S.H.P. . . | 217·3 | 211·5 | 201·3 | 211·3 | 227·3 |

* H.M.S. *Bristol* has twin screws.

(6) S.S. *Reina Victoria Eugenia* is a four-screw passenger-cargo steamer, 480 ft. × 61 ft. × 35·8 ft. Of the four screws, the outer pair are each driven by a separate low-pressure turbine, 95 inches diameter, taking steam from the corresponding reciprocator triple driving one of the inner pair of screws, and having cylinders 29 inches and 43 inches, and two low-pressure 47 inches diameter by 42 inches stroke. There were two crucial trials, one deep laden to 24·7 feet mean draught, and the other on 19·85 feet, as she would be loaded with light cargo or partly laden with heavy goods.

## Trials of 4-S.S. "Reina Victoria Eugenia."

| | Fully Laden. | Part Laden. |
|---|---|---|
| Displacement . . . . . . tons | 13,229 | 10,181 |
| Horse-power developed . . . I.H.P. | 5,760 | 7,340 |
| S.H.P. | 2,157 | 3,500 |
| total H.P. | 7,917 | 10,840 |
| Revolutions per minute . . reciprocators | 102·9 | 112·6 |
| turbines | 395·0 | 481·0 |
| Speed mean . . . . . . knots | 16·10 | 18·12 |
| Slip per cent. . . propellers of recipros. | 4·0 | 1·0 |
| ,, turbines | 13·0 | 20·0 |
| Admiralty co-efficient $D^{2/3} \times S^3 \div$ I.H.P. . . | 294 | 257 |
| Consumption of steam per I.H.P. hour . total | 12·375 | 15·17 |
| ,, ,, ,, only main engines . . . . . . . . . | 10·625 | ... |
| Steam pressure at valve chests of recipros. lbs. | 170·50 | 170·00 |
| ,, ,, turbine entrance . . lbs. | 5·40 | 7·80 |
| ,, ,, condenser . . . lbs. | 0·56 | 0·50 |
| Temperature of sea-water . . . . . . | 45° F. | 43° F. |
| ,, discharge water . . . | 64° F. | 70° F. |
| ,, hot well . . . . . | 62° F. | 70° F. |

Steam was supplied by seven single-ended boilers, 16 feet 3 inches diameter × 11 feet 6 inches long, having a total heating surface of 20,965 square feet so that 5·67 lbs. of steam per hour were given by each square foot of T.H.Surf., with an air pressure of 0·35 inch in the ash-pits with Howden's system of draught.

(7) S.S. *Cairncross* is 370 ft. × 51 ft. × 27·8 ft., of 9950 tons displacement on 23·4 feet mean draught, the block coefficient 0·779. She has a single screw of large diameter, driven by two (high-pressure and low-pressure) turbines geared by pinions and spur wheel. Steam is supplied from three boilers at a pressure of 180 lbs. Her normal speed in fine weather is 10 knots.

Her sister ship, the *Cairngowan*, has similar boilers and propellers, but driven by triple compound engines, with cylinders 24 inches, 40 inches, and 66 inches diameter and 45 inches stroke. They were laden practically to the same displacement, supplied with the same coal, and steamed side by side for 36 hours. The weather, however, was bad, so that no account is made of the speed.

It will be seen that the turbine proved a more economic instrument than the reciprocator, and was worked with a vacuum of 28·75 inches as against 26·8 inches in the *Cairngowan.* No doubt, too, that in rough seas the turbine-driven screw did not race so much as the other, and thereby got drier steam, etc.

The ratio of spur wheel to pinions is 26.

## Special 36 Hours' Sea Trials of S.S.'s "Cairncross" and "Cairngowan."

| | *S.S. Cairncross.* | *S.S. Cairngowan.* |
|---|---|---|
| Machinery, kind of . . . . | Geared Turbines | Triple Compound |
| Mean revolutions of screw, p.m. . | 61·76 | 61·68 |
| Horse-power developed . . . | 1570 S.H.P. | 1790 I.H.P. |
| Steam pressure at engines . . | 138 lbs. | .. |
| Temperature of hot well . . F.° | 79° | 104° |
| Vacuum in condenser . . ins. | 28·75 | 26·80 |
| Coal consumed per 24 hours . tons | 27·8 | 32·7 |
| ,, ,, ,, I.H.P. hour lbs. | =1·45 | 1·704 |
| Coal per sq. ft. of grate . . ,, | 17·9 | 21·0 |
| Water per hour . . . . ,, | 22,400 | 27,200 |
| ,, ,, I.H.P. hour . . ,, | =12·57 | 15·18 |
| Ash from coals . . per cent. | 8·50 | 8·97 |
| Assuming calorific value 14,500 B.T.U., the efficiency of system is | 12·10 | 10·32 |
| Saving of fuel . . . per cent. | 15·0 | .. |

## Actual Steam Consumption on Two Large Liners.

| Description of Machinery. | S. H. P. | Steam at Turbines. | Consumption in Lbs. per Hour. | | |
|---|---|---|---|---|---|
| | | | Main Turbines. | Auxiliaries and other Purposes. | Total for all Purposes. |
| 4 Shafts direct-coupled turbines . . . | 23,000 | Saturated | 11·2 | 2·2 | 13·4 |
| 2 Shafts geared compound turbines, 1 H.P. and 1 L.P. to each. | 14,500 | 85° F. superheat. | 9·7 | 2·3 | 12·0 |

## Methods of Transmission from Turbine to Screw Compared. (Cleghorn.)

| Points of Comparison in a Ship of 20,000 S.H.P. | Method of Transmission. | | |
|---|---|---|---|
| | Double Wheel Gearing. | Hydraulic Transformer. | Geared Electrical Generator, and Motor Driven. |
| Rev. of turbines per min. . | 2400 and 1500 | 1000 (about) | 2400 and 1500 |
| ,, screws ,, . | 90 | 140 | 90 |
| Transmission efficiency . . | 0·965 | 0·92 | 0·885 |
| Relative fuel consumption | 100 | 123 | 109 |
| ,, machinery weights | 100 | 109 | 108 |

# INTERNAL COMBUSTION ENGINES.

These engines are such as consume their fuel in the same cylinder or other vessel in which the energy is imparted to the mechanical means of motion. The fuel may be in the gaseous state produced from coal, oil, or other volatile substances giving off hydrocarbon, or it may be the spray from oils injected into the cylinder.

**Gas engines** on board ship have not proved an unqualified success, and since they are generally supplied from producers, which are heavy and occupy much space, they are at a considerable disadvantage in competition with oil engines.

**Oil engines** may be divided into three classes:—

(*a*) Those using petrol or other *light* volatile oil.
(*b*) Those working with paraffin, having a higher flash point.
(*c*) Those working with heavy oils, having such a high flash point as to be perfectly safe on shipboard.

**Petrol**, so much in demand for locomotion on land, is too dangerous and now too dear for maritime purpose generally. It is convenient, on account of its vapourising so readily, for starting engines whose general fuel is heavy oil. With a flash point of 70° F. it is also used on small pleasure boats, where very high speed is required. It is too dangerous for handling in confined engine-rooms or on shipboard at all.

**Paraffin**, which is cheaper than petrol, can be got anywhere, has a flash point of 90° to 120° F., which renders it safe to handle, and, as may be seen by Table LXXI., it has high thermal value, quite as high as petrol. Paraffin engines are largely employed on small craft for propulsion with success, and the demand for them, since petrol is becoming so dear, is likely to increase. The flash point of paraffin is, however, too low for the Admiralty, whose limit is 175° F., as it is also for Lloyd's, whose limit is 150° F.

**Alcohol**, which can be produced so cheaply everywhere, has not yet been used seriously for propulsion inasmuch as the flash point is high and uncertain, so that with a Diesel or Semi-Diesel engine it misses fire often and cannot be relied on. Its calorific value is much lower than that of the petroleum products, being only 10,260 B.T.U. per pound of 90 per cent. mixture, or not much more than half that of good heavy oil.

**Heavy oils**, with flash points from 175° to 250°, are largely used now on shipboard as well as on shore. They are mostly residuals of kinds, even when called crude oils; in fact, unless an oil has undergone some sort of a refining process, it is not fit for an oil engine to use. Some residuals are left after all the lighting and lubricating oils have been abstracted. Some are crude oils after some exposure to the air to get rid of sulphur and dangerous volatile elements. In some so-called crude oils there remains an amount of bituminous matter which gets converted into hard sandy coke; this asphaltum really spoils the oil for internal combustion purposes, as the coke formed from it will not fuse or consume away, but remains in a crystalline state fit to cut the pistons, valves, &c.

## Table LXXI.—Oils and other Liquids used as Fuel in Internal Combustion Engines.

| Name of Liquid. | Specific Gravity. | Flash Point. | Boiling Point. | Thermal Value per lb. | Composition. | |
|---|---|---|---|---|---|---|
| | | | | | Hydrogen. | Carbon. |
| | | F.° | F.° | B.T.U. | | |
| Light Oils in general | 0·65 to 0·76 | 70 | 300 | .. | .. | .. |
| Gasoline | 0·66 to 0·67 | .. | 158 to 176 | .. | .. | .. |
| Benzine | 0·68 to 0·75 | .. | 176 to 302 | .. | .. | .. |
| Petrol in general | 0·66 to 0·72 | .. | 158 to 230 | 18,360 | 14 to 16 | 86 to 84 |
| Heavy Oil, medium | 0·76 to 0·86 | 70 to 158 | 300 | 20,000 to 18,000 | 13·6 to 14·8 | 86·3 to 82 |
| Paraffin, Russian and American | ,, | ,, | ,, | 19,000 | .. | .. |
| ,, Russian (best) | 0·82 to 0·83 | 90 | .. | 19,000 | .. | .. |
| Standard Oil | 0·81 | 84 | .. | 19,000 | .. | .. |
| Heavy Oils in general | 0·86 to 0·94 | 140 to 270 | .. | 18,800 | 12·3 to 13·7 | 86·6 to 83·5 |
| Solar Oil | 0·86 to 0·89 | 140 to 270 | .. | 17,400 to 18,600 | .. | .. |
| Texas Oil | 0·91 to 0·95 | 130 to 270 | .. | 18,000 to 18,900 | 10·9 | 84·6 |
| Galician Solar Oil | 0·86 to 0·88 | . | .. | 18,500 | 12·6 | 85·3 |
| Borneo Oils | 0·91 to 0·95 | 176 | .. | 17,600 | .. | .. |
| Alcohol Methyl | 0·834 | .. | .. | 10,300 | 12 | 43 to 45 |
| ,, and Benzol, mixed | .. | .. | .. | 13,700 | .. | .. |
| ,, American, denatured | 0·8199 | .. | .. | 10,512 | 12·68 | 47·04 |
| Tar Oil, distillate | 0·92 | .. | .. | 17,500 | .. | .. |
| Benzol | 0·882 to ·885 | .. | 175 | 17,100 | .. | .. |
| Shale Oil, Swedish | 0·997 | 130 | .. | 17,487 | 9 | 87·8 |
| ,, French | 0·911 | .. | .. | 18,700 | 11·5 | 80·3 |
| ,, Scotch | 0·86 | 220 | .. | 16,775 | 7·0 | 86·5 |
| Creosote Oil | .. | 400 | .. | 15,800 | 7·6 | 82 |
| Mexican Crude | 0·950 | over 150 | .. | 18,800 | 11·68 | 83·52 |

*N.B.—The British Admiralty* require that the oil fuel admitted on board His Majesty's ships shall have a flash point no lower than 175° F., and shall contain no more than 0·75 per cent. of sulphur.

*The weight of air required* for the combustion of petrol and paraffin oil is 184 times that of the oil; for heavy oils, as used on Diesel engines and the residuals in boilers, 176 times is sufficient. Benzol requires only 152, and alcohol (90 per cent.) will do with 96 times. Petrol generally has a flash point over 80° F. and Paraffin over 120° F., as supplied for use in engines.

**Creosote, or tar oil,** is a home product which can be used in these engines quite well, especially after a little primitive refining; the calorific value is 15,800 B.T.U. per pound.

**Shale oil,** as distilled from the shale in Scotland, is also a valuable one for fuel and easier to run with than the petroleum residuals; it is freed from naphtha spirit and paraffin wax, and then has a calorific value of 16,775 B.T.U. and a flash point of 220° F.

In course of time other liquid fuels will be found which may be used in oil engines, for already the late Dr Diesel had tried successfully some vegetable oils.

Petroleum, as found in various parts of the world, differs somewhat in chemical composition and characteristics; they are sometimes classed as *Asphaltic* and *Paraffin* oils in America; the Russian and East Indian oils can hardly be so classed, however.

**Asphaltic oils** are of high specific gravity (0·945), high in sulphur, harden under atmospheric influences, and their residues tend to solidify; they are subject to decomposition at comparatively low temperatures, and their distillates have a high solvent power.

**Paraffin oils** are of lower specific gravity (0·832), are low in sulphur, are not affected by exposure to air, have good capillary power, and their distillates have only low solvent properties.

The sulphur in the Californian (Asphaltic) oils is as much as 3·3 per cent., as against only 0·3 per cent. in Texas oil. The British Admiralty used to require their oil supplies to have no more than 0·75 per cent. of sulphur and the flash point to be not lower than 200° F.; now 3 per cent. of sulphur and 175° are permitted. The boiling point of such oil is about 300° F.

**Oil engines are subject to another classification,** viz:—

(1) Those which draw in the charge of fuel with the air and compress the mixed charge lightly, say to 80 lbs. pressure per square inch, before explosion.

(2) Those which draw in a charge of air only and compress it moderately, say to about 150 lbs., or about 60 per cent. of the pressure produced by the explosion of a charge of oil vapour sprayed on to a hot plate or sphere. This kind is known as the *Semi-Diesel.*

(3) Those which draw in a charge of air and compress it so highly (400 to 500 lbs.) that it becomes sufficiently hot to ignite the charge of oil spray on entering it and produce an explosion without shock.

This system, wherein the air is compressed by itself alone to a temperature sufficient to ignite the charge of oil spray and to burn it without any serious increase to the pressure, was invented and perfected by the late Dr Diesel, and is known as the Diesel System.

There is still another differentiation among oil engines, and now, for marine engineers, an important one.

Both the *Diesel* and the *Semi-Diesel* oil engines may be worked on either the two-stroke or the four-stroke cycle. The latter is the older and known as the "Otto"; most gas and land oil engines work on it, and altogether it is simple and reliable; but it involves larger or more numerous cylinders for the same power, consequently for marine propulsion the two-stroke cycle is the one in general use.

**In the Otto cycle**, with a single-acting engine, the piston makes an outward stroke, and thereby sucks into the cylinder a mixture of gas and air in the ordinary engine and a charge of air only in the Diesel engine. On the second, or return stroke, there is compression more or less, as already named; ignition takes place, so that this, the third, is the active stroke; on the fourth, or return stroke, the products of combustion contained in the cylinder are driven out, and it is called the scavenging stroke.

**In the two-stroke cycle** the cylinder is scavenged and filled with fresh air, so that the first stroke of the piston compresses it; at the commencement of the second stroke fuel is sprayed in, ignited, and burnt during that stroke; at the end fresh air is again caused to scavenge and charge. In this way there is an active stroke every revolution, while in the Otto there is one only for every two revolutions.

**The double-acting oil engine**, as the name signifies, is active on both sides of the piston, and can, as a consequence, have smaller cylinders still. So far, however, this kind of engine has been successful only in small high-speed craft, but it is said the German Naval ships of large size are being so fitted. The Augsburg-Nürnberg Co., however, have made, and are still engaged in experiment with and perfecting of, engines of this kind of considerable size for Naval purposes.

For obvious reasons, with such very high initial pressures and high temperatures, cylinders of very large diameter are almost impossible and certainly undesirable. Consequently, for large power, a large number is usually found.

**The number of cylinders** in marine oil engines, for this cause and as a consequence of the difficulty of keeping the engine running steadily at low revolutions, must be numerous; and even when there are four cylinders a heavy fly-wheel is really necessary for safe running "dead slow." There are seldom fewer than four cylinders to marine oil engines, and often double that number when the power is large, so that in twin-screw ships there are sixteen cylinders, and even in quite small ships as many as twelve. The four-cycle engine, single acting, should, of course, never have less than four cylinders on a single line of shaft, whatever the power may be, and even then a fly-wheel is necessary for steady running.

In a general way it may be assumed that, whatever type of oil engine is used, the larger the number of cylinders the better it runs; the torque is more uniform, and the certainty of continuous running at "dead slow" is ensured. On the other hand, the larger the number the more numerous are the working parts all requiring attention, lubrication, and power for driving.

No marine oil engine of the two-cycle type should have less than three cylinders, even if double acting; and all single-acting ones should have not less than four. For an even torque six cylinders are best, and then the fly-wheel can be dispensed with.

For very large powers six and eight cylinders must be employed on each line of shafting. It is convenient in such cases to double the engine into two sets of three or of four cylinders, each capable of running independently, so that with a clutch or other good coupling between them it can be arranged to run only one half when for cruising or other purpose low power only is required.

It should be noted that the efficiency of the two-stroke cycle is 2 per cent. less than that of the four-stroke, for a separate compressor gives a loss of 4 per cent.

Although the fuel of the Diesel and Semi-Diesel is a safe heavy oil, these engines will not start with it, so that a small stock of a more volatile oil is necessary for this purpose, as well as to clean out with just before stopping if some hours are to elapse before restarting.

The paraffin engine also requires, as a rule, a charge or two of petrol or ether to start with.

Any oil engine will work readily ahead or astern if it can be started into motion and ignition caused to take place just as the piston commences the down stroke. The starting and manœuvring of such engines, which proved difficult in the past with even small sizes, are now effected easily and quickly by means of compressed air.

**The reversing of the propeller** has been accomplished in small craft by such means as turning the blades through a considerable angle with the engine running, or by means of wheel gearing, as in the motor car.

In the large engine of to-day there is always the means of starting and reversing provided, whereby air compressed to very high pressure is admitted to the oil cylinders by means of valves operated by gear which is reversible, as in a steam engine ; and it is claimed that the large oil engine in the hands of trained engineers is as handy and reliable for manœuvring as any steam engine.

**The general design** of the marine oil engine varies somewhat ; in the small sizes there is usually the enclosed casing surmounting the bed-plate and supporting the cylinders, besides providing guides, etc., as introduced by Belliss & Morcom for their high-speed electric generating engines with forced lubrication, &c. The larger engines are designed generally on the lines of the marine vertical steam engine, with such modifications as are necessary for oil-engine appliances.

**The cylinders** have a piston stroke longer in proportion to the diameter than usual with steam ; their "heads" and covers have to be very strongly constructed and secured to resist shock ; in design and material generally the greatest care is necessary, having regard to the high temperature as well as high pressure to which they are exposed. Water-jacketing is resorted to freely, and seeing that 40 per cent. of the heat is carried away by the cooling water, the circulation must be free.

**The pistons** are always fitted with Ramsbottom spring rings freely distributed to prevent leakage at all times. Water-cooling is resorted to here in big engines, and in double-acting engines of all sizes cooled pistons are necessary. In the two-cycle engine the scavenging pump is often immediately below the piston, so that a piston-rod is sometimes necessary to make the connection with the connecting-rod. Generally speaking, a piston-rod is necessary and always desirable to keep the connecting-rod top end in a cool place. It is certainly not desirable to have it in a trunk close to the piston face.

**The guides and shoes** may be just as in a steam engine and follow the same rules for surface, but if the initial pressures are taken for purposes of calculation a higher nominal pressure per square inch can be adopted.

**Rule 224. Gross area of guide shoe** $=\frac{T \times \sqrt{S+100}}{3050}$ sq. ins.

$T =$ area piston × initial pressure ÷ 5,
$S =$ piston speed in feet per minute.

The crankshafts are subject to considerable straining from the intermittent action of the pistons as well as the great range of pressure —the maximum being over 5 times the mean.

**Shafts of Oil Engines.**—Lloyd's Rules when shafts are made of ordinary mild steel. If special steels are used, the size must be submitted for consideration.

**Rule 225. 1. Diameter of crankshaft in inches** $= C\sqrt[3]{D^2 \times S}$.

D is the diameter of cylinder and S the stroke of piston, each in inches. C is a multiple of value as follows for smooth-water service :—

| Four-Stroke Cycle. | Two-Stroke Cycle. | Bearing between each Crank. | Two Cranks between the Bearings. |
|---|---|---|---|
| For 1, 2, 3, or 4 Cylinders | 1 or 2 Cylinders | C=0·34 | C=0·38 |
| ,, 6 ,, | 3 ,, | C=0·36 | C=0·40 |
| ,, 8 ,, | 4 ,, | C=0·38 | C=0·425 |
| ,, 12 ,, | 6 ,, | C=0·44 | C=0·49 |

For the Open Sea Service add 0·02 to each value.

**Rule 226. Diameter of screw shafts in ins.** $= G\sqrt[3]{D^2 \times S(N+3)}$.

N is the number of cylinders ; if two-stroke, N = twice the number. The following are the values of G :—

| | Smooth Water Service. | Open Sea Service. |
|---|---|---|
| For all intermediate shafts, . . . | G=0·155 | 0·165 |
| ,, screw shafts with continuous liners, . | G=0·170 | 0·180 |
| ,, ,, ,, separate or no liners, | G=0·180 | 0·190 |

With large thrust collars the shaft between them must be $^{21}/_{20}$th the diameter of shaft elsewhere. For Diesel engines and others with high initial pressures, sizes must be submitted.

**Rules of the Bureau Veritas** for the shafting of Internal Combustion Engines.

**Rule 227. Diameter crankshaft in ins.** $= 0{\cdot}106\sqrt[3]{G \times D^2 \times S}$.

$D$ = diameter of cylinder in inches,
$S$ = stroke of piston ,,
$G$ = a variable factor.

### Values of G for a Ratio $\frac{S}{D}$.

| $\frac{S}{D}$ | Four-Cycle Single-Acting Engines. | | | | |
|---|---|---|---|---|---|
| | 6 Cylinders. | 8 Cylinders. | 10 Cylinders. | 12 Cylinders. | 16 Cylinders. |
| | Two-Cycle Single-Acting Engines. | | | | |
| | 3 Cylinders. | 4 Cylinders. | 5 Cylinders. | 6 Cylinders. | 8 Cylinders. |
| | Two-Cycle Double-Acting Engines. | | | | |
| | | 2 Cylinders. | | 3 Cylinders. | 4 Cylinders. |
| 1·0 | 127·867 | 129·849 | 135·082 | 139·046 | 151·657 |
| 1·1 | 119·224 | 121·336 | 126·894 | 130·989 | 144·184 |
| 1·2 | 112·221 | 114·431 | 120·281 | 124·506 | 138·156 |
| 1·3 | 106·425 | 108·700 | 114·745 | 119·165 | 133·140 |
| 1·4 | 101·526 | 103·931 | 110·139 | 114·714 | 129·021 |
| 1·5 | 99·421 | 99·891 | 106·261 | 111·331 | 125·501 |
| 1·6 | 93·863 | 96·365 | 102·943 | 107·708 | 122·528 |
| 1·8 | 88·139 | 90·739 | 97·564 | 102·439 | 117·714 |
| 2·0 | 83·693 | 86·422 | 93·430 | 98·480 | 114·015 |

**Rule 228. The diameter of the intermediate shafts,** $d_1 = n\sqrt[3]{D^2 \times S}$. The values of the factor $n$ are given in the following table:—

| Four-Cycle Single-Acting. | Two-Cycle Single-Acting. | Two-Cycle Double-Acting. | $n$. |
|---|---|---|---|
| 6 Cycles | 3 Cycles | .. | 0·389 |
| 8 ,, | 4 ,, | 2 Cycles | 0·397 |
| 10 ,, | 5 ,, | .. | 0·415 |
| 12 ,, | 6 ,, | 3 Cycles | 0·427 |
| 16 ,, | 8 ,, | 4 ,, | 0·459 |

**Rule 228a. The diameter of the thrust shaft,** in way of thrust block, $d_2 = d_1 + 6{\cdot}4$ per cent.

**Rule 229. The diameter of the propeller shaft,** $d_3 = d_1 + \frac{D}{100}$, where $D$ is the diameter of the propeller in inches.

**In estimating the fuel** consumed per unit of power transmitted for use, full account must be taken of *all fuel* used in auxiliaries; moreover, as these engines require a very large amount of lubricating oil, it would be only fair to include it as fuel in both the oil and steam engine. In both cases the consumption (total) in pounds per hour should be divided by the Brake Horse-power, or Shaft Horse-power.

**The auxiliaries** or additions required by the two-stroke cycle marine engines are (i) a scavenging pump delivering air direct to the oil cylinders, (ii) a pump for compressing and storing the air necessary for manœuvring the engines, (iii) another compressor for the air of very high pressure for feeding and spraying the fuel, (iv) a pump to circulate the cooling water; (v) a pump to deal with the fuel.

(i) The scavenging pump is generally an ordinary single-acting air pump worked by the main engine, and having a capacity from 1·25 to 1·35 that of the cylinders supplied.

(ii) The pump for the supply of air for handling the engine is also generally worked by the main engine and delivers at a pressure of about 300 lbs. per square inch, for which purpose it has to be of the two-stage type. This pump, however, has been lately arranged to be worked by an independent engine.

(iii) The air for oil supply is of higher pressure—about 750 lbs.; it is taken from the 300-lb. reservoirs and further compressed by a small pump worked from the main engine.

**The fuel consumption** of the Diesel engine of the two-stroke cycle type is generally about 0·45 lb. per B.H.P. hour; the mechanical efficiency of these of good size is about 0·78; so that the consumption per I.H.P. is 0·351 lb. Some large engines when new and on trial with skilled hands have done better, viz., 0·38 lb. per B.H.P. or 0·300 lb. per I.H.P. hour. Generally speaking, half a pound of good heavy oil (calorific value 19,000 B.T.U. per lb.) per B.H.P. is a good and satisfactory result in everyday work.

When this is done the efficiency (thermal) is 0·343 (*v.* Rule 17). Tested in this way, the turbine consuming 12 lbs. of steam would require to use 0·70 lb. of equally good fuel and the efficiency would be 0·191. The ordinary turbine with a consumption of 13·5 lbs. would show only 0·167, or less than half that of the oil engine.

The Diesel engine's consumption per B.H.P. at lower powers does not increase so rapidly as does that of steam engines as may be seen by the following table.

### Table LXXII.—Consumption of Oil at Various Loads per B.H.P. Hour in lbs.

| Load. | 400 B.H.P. Engine. | | 200 B.H.P. Engine. | |
|---|---|---|---|---|
| | Crude Oil (Sulzer). | Crude Oil (Chalkley). | Creosote Oil. | Crude Oil. |
| Full power, . . | 0·471 | 0·473 | 0·518 | 0·442 |
| Three-quarters load, . | 0·471 | 0·475 | 0·462 | 0·462 |
| Half load, . . | 0·477 | 0·496 | 0·528 | 0·506 |
| One-quarter load, . | 0·546 | 0·493 | | |

The following are the results of some trials made with an oil engine having cylinders 6·5 inches diameter, 10·6 inches stroke, running at 280 revolutions per minute, with various fuels whose composition and thermal values are as follow:—

### Table LXXIIa.—Particulars of Various Oils tried in this Engine.

| Designation of Fuels. | Specific Gravity. | Lower Heat Value, B.T.U. | Per cent. of Carbon. | Per cent. of Hydrogen. |
|---|---|---|---|---|
| Royal Daylight paraffin, | 0·811 | 18,500 | 85·0 | 15·0 |
| Pratt's petrol, . . | 0·703 | 18,730 | 84·0 | 16·0 |
| Texas heavy oil, . . | 0·882 | 18,190 | 86·3 | 13·7 |
| Common paraffin, . | 0·815 | 18,650 | 85·0 | 15·0 |

The following is an abstract of some of the trial results.

### Table LXXIIb.—Results of Trials of this Oil Engine with the above Fuels.

| Kind of Fuel used. | Fuel per B.H.P. Hour. | B.H.P. | I.H.P. | Thermal Efficiency. | Mechanical Efficiency. | Total Efficiency. | Heat carried away | | Temperature at Exhaust. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | By Water. | Exhaust. | |
| | lb. | | | | | | | | F.° |
| Texas heavy oil, | 0·540 | 7·95 | 11·90 | 38·80 | 67·0 | 25·80 | 32·7 | 28·5 | 477 |
| ,, ,, | 0·521 | 5·57 | ... | ... | ... | 26·80 | 31·5 | ... | 383 |
| Royal Daylight paraffin, . | 0·624 | 7·92 | 10·26 | 30·25 | 72·4 | 21·80 | 29·5 | 40·2 | 455 |
| Pratt's petrol, . | 0·743 | 6·87 | ... | ... | ... | 18·25 | 28·3 | ... | 562 |
| Common paraffin, | 0·489 | 8·55 | ... | ... | ... | 27·80 | ... | ... | ... |
| ,, ,, | 0·507 | 5·60 | ... | ... | ... | 26·9 | 31·5 | ... | ... |

**The mechanical efficiency** of the marine oil engine is not so high as that of a steam engine of equal power, as the friction, both internal and external, is greater than that of a quadruple compound, besides which there are the various pumps driven by the main engine which require more power than the air and feed of the steam one. It is probable that with the large slow-moving engines of the mercantile marine the mechanical efficiency is under 78 per cent., and all of this class of engine show poor efficiency at lower speeds, so that when cruising the general efficiency will not be so very superior to the reciprocating steam engine.

**The mean pressure** in the marine oil engine at full power is from 95 to 115 lbs. per square inch. In special cases it is 125 lbs., and efforts are being made to increase this for very high-speed warship engines in Germany. Generally for good and economic working 100 to 110 lbs. is a fair allowance for estimating purposes.

**The rate of revolution** can be more or less arbitrarily decided on; as a rule, it is higher size for size than prevails in steam engineering in the mercantile marine.

*Burmeister & Wain*, with their associated friends, have a kind of standard rate which varies from 250 revolutions per minute with single-cylinder 30 H.P. to 150 revolutions for 1000 H.P. four-cylinder engines. All these engines from 40 to 250 H.P. may have two cylinders; from 50 to 750 H.P. they supply also three-cylinder ones; while they make four-cylinder engines of 50 H.P., which they run at 225 revolutions and up to 1000 H.P. at 150 revolutions.

*The Augsburg-Nürnberg Co.* design their 36 H.P. engines for 250 revolutions with two cylinders, and make four-cylinder 200 H.P. for 195 revolutions, while their 1000 H.P. are for 150 revolutions.

*Sulzers* will make engines of 8000 H.P. to run at 105 revolutions. Naval engines are designed for higher rates, so that the 1200-H.P. engine is made for 400 revolutions.

The *Selandia* of 2500 I.H.P. has engines running at 140 revolutions; each engine is, of course, 1250 H.P. The *Eavestone* was designed for and has run at 90 revolutions, indicating 1370 I.H.P.

The following table gives the rates of revolution generally aimed at by the Augsburg-Nürnberg Co.

### Table LXXIII.—Rates of Revolution of Augsburg-Nürnberg Oil Engines.

| Size of Engine. | Revolutions per minute. | |
|---|---|---|
| | Light Naval. | Heavy Type. |
| 150 horse-power, . . | 550 | 400 |
| 200 ,, . . | 550 | 330 |
| 300 to 500 horse-power, . | 500 | 330 to 275 |
| 600 horse-power, . . | 450 | 275 |
| 900 ,, . . | 420 | 260 |
| 1200 ,, . . | 400 | 215 |

The rate of revolution of marine engines may be determined by the following rules, and be in accordance with the above and other good practice:—

**Rule 230.** Number of revs. per min. $= 2300 \div \sqrt[4]{\text{H.P.}}$ Light naval.

**Rule 230a.** ,, ,, ,, $= 2300 \div \sqrt[3]{\text{H.P.}}$ Heavy ,,

**Rule 230b.** ,, ,, ,, $= 1500 \div \sqrt[3]{\text{H.P.}}$ Mercantile.

**The weight of the oil engine** is considerable per H.P., so that an oil-engine installation is often as heavy as a steam one.

(*a*) The very lightest, such as made specially for small Naval ships, weigh 40 lbs. per S.H.P. of the smaller and 35 lbs. of the larger powers, as against 43 lbs. of reciprocator destroyers.

(*b*) The ordinary mercantile installation weighs from 300 lbs. of small to 215 lbs. of large engines per S.H.P. The slower speed ones, such as the *Eavestone*, are heavier still.

(*c*) The engines made for express short-passage ships are, of course, considerably lighter, being from 90 to 100 lbs. per I.H.P. The machinery of similar ships with reciprocators would be 210 lbs. per I.H.P., and turbine installations 123 lbs.; while that of a third-class cruiser with water-tube boiler is 120 lbs. with reciprocators, and 95 with turbines.

**The space occupied by oil-engine** installations is very much less than that by steam ones, and practically by the amount of the boiler room, for even the 8-cylinder engines require little or no more space than the 4-crank engines and condensers of the steamship. The space occupied by oil fuel as against that by coal is a most important factor. Leaving aside the question of taxing the space in the same way as coal bunkers, there remains the fact that oil can be stowed at little cost for labour in spaces which are inaccessible for cargo, and brought from them to the engines and boilers at equally small cost. The application of the fuel requires no hand labour and practically costs nothing, so that altogether the labour costs in an oil-fuel ship are very small indeed. Besides which, the weight of the firemen not required, together with that of the food, water, and appliances for them, which will be considerable, is saved and in its place freight-paying cargo is carried. On the other hand, it must not be forgotten that whereas any kind of fuel can be used in a boiler, only certain kinds and qualities of *oil* can be used in an oil engine; so that in case of accident the oil is lost, a further supply is not to be got, and the oil-engine ship is helpless. Moreover, in this country we have an abundant supply of coal, but we have little or no oil—nor have we as yet an oil-field near at hand in foreign countries. As long as crude oil could be got at 40 to 50 shillings a ton it was worth while having an oil engine here, especially for intermittent work, but the demand before the war had caused it to rise to over 90 shillings; it is now more costly than coal with a steam engine, and likely to be still higher if the mercantile marine goes in largely for this class of engine, as this would expand the demand enormously; but doubtless other oil-fields will be found.

The advantages of the Diesel system over others are:—

(1) Safety due to the use of a harmless high flash-point fuel.

(2) High compression of air free from fuel, whereby :—
- (*a*) The use of an ignitor is avoided.
- (*b*) Explosion, or combustion rather, is effected without shock.
- (*c*) The mixture of fuel with the air is more certain.
- (*d*) A poorer mixture can be used when desirable.
- (*e*) There is no fear of pre-ignition or back-firing.

(3) Less cooling water is required consequent on the less fuel consumed.

The Semi-Diesel engine possesses most of these advantages, but it requires an ignitor, which is liable to fail in action when dirty.

Of the heat generated by the combustion of the fuel, 38 to 40 per cent. is converted into work, 40 per cent. is absorbed by the cooling water or lost by radiation, and 20 per cent. passes away at the exhaust. A considerable amount of the 60 per cent. otherwise wasted can be recovered by means of a steam boiler, as the temperature at exhaust is about 450° F.—that is, sufficiently high to evaporate some of the water from the jackets which has been raised to near the boiling point.

The steam from this auxiliary boiler can be employed for domestic purposes as also for steering and driving the auxiliary machinery generally.

It means that if in the *Eavestone* 33 per cent. of the loss can be recovered, there should be produced 1000 lbs. of steam per hour; enough to supply auxiliaries of an aggregate of 46 I.H.P.

**The indicated horse-power** can be found as follows :—

D is the diameter of the cylinders in inches whose number is $n$;
S is the stroke of piston in feet ; $p_m$ is the mean pressure ;
R is the number of revolutions per minute.

**Rule 231.**

$$\left.\begin{array}{r} \textbf{I.H.P.} = \dfrac{D^2 \times S \times p_m \times R}{25{,}338} \times n \\[2ex] \textbf{Diam. of cylinder} = \sqrt{\dfrac{\text{I.H.P.} \times 25{,}338}{S \times p_m \times R \times n}} \end{array}\right\} \text{for 4-stroke cycle single-acting.}$$

**Rule 232.**

$$\left.\begin{array}{r} \textbf{I.H.P.} = \dfrac{D^2 \times S \times p_m \times R}{12{,}669} \times n \\[2ex] \textbf{Diam. of cylinder} = \sqrt{\dfrac{\text{I.H.P.} \times 12{,}669}{S \times p_m \times R \times n}} \end{array}\right\} \text{for 2-stroke cycle single-acting.}$$

**Rule 233.**

$$\left.\begin{array}{r} \textbf{I.H.P.} = \dfrac{D^2 \times S \times p_m \times R}{6335} \times n \\[2ex] \textbf{Diam. of cylinder} = \sqrt{\dfrac{\text{I.H.P.} \times 6335}{S \times p_m \times R \times n}} \end{array}\right\} \text{for 2-stroke cycle double-acting.}$$

**Limitation to diameter of cylinder** is enforced by one or two practical considerations ; the thickness will increase with the diameter and soon become so great as to seriously retard the passage of heat to the cooling water, and the strength of the shell to resist internal pressure and shock will not be proportional to the thickness. The surface exposed to heat increases as the diameter, while the quantity of heat will vary as the square of the diameter, so that the ability to keep sufficiently cool decreases as the diameter increases. In practice with the requirements of the mercantile marine, the diameter seldom exceeds 24 inches, and probably 30 inches is the safe limit with cast-iron. With the practice of Naval engineering a somewhat larger cylinder is possible, especially for ships like scouts and cross-channel expresses, where a run at fullest speed is a matter of minutes; the German firms have made 30-inch cylinders for such craft, and Burmeister for one merchant ship.

The thickness of cylinder is, of course, much greater for merchant ships than for fast expresses and naval craft.

**Rule 234. Thickness of cylinder,** $= D \times f$ for Diesel engines.
For merchant ships, $f = 0·1$ to $0·08$.

$$\left.\begin{array}{r}\text{Naval and expresses, } f = 0{\cdot}06 \\ \text{Very light engines, } f = 0{\cdot}05\end{array}\right\} \text{ of superior iron.}$$

The cylinders are often made thicker at the end where there is greatest compression and the explosion; then the thickness tapers and varies from $f = 0·10$ to $f = 0·06$.

The cylinders of Semi-Diesel engines may be, of course, thinner, so that $f$ is $= 0·6$ of the above values of $f$.

This means that the metal, which must always be of a special kind, is nominally stressed to 2750 lbs. per square inch in mercantile engines, and as much as 5000 lbs. in naval high-speed engines.

The following are a few examples of the performance of the heavy oil engine in actual practice.

(1) *A Diesel engine* with cylinders 22·05 inches diameter × 29·5 inch stroke running at 150 revolutions per minute.

The consumption of oil fuel per hour was 195·2 lbs.
The maximum pressure in cylinders, 525 lbs. per square inch.
The mean ,, ,, 95 ,, ,,
The indicated horse-power—full speed, 609; slow speed, 163·3.
The brake ,, ,, 475·5; ,, 54·6.
Fuel per B.H.P. hour, lbs. ,, 0·411; ,, 0·838.
Efficiency, B.H.P. ÷ I.H.P. ,, 0·78; ,, 0·334.

The pumps were driven by a special engine whose efficiency was 0·715 and the horse-power 44·8. Taking into account the fuel expended on these engines and the B.H.P. of the main engine only, the consumption of fuel was 0·444 full power, 1·415 lbs. low power.

(2) *The S.S. Eavestone* had Diesel engines (Westgarth Carel) with four cylinders 20 inches diameter, 36 inches stroke in which a mean pressure of 127 lbs. has been obtained. She attained a rate of 9 knots, loaded on a consumption of 3·5 tons of crude oil fuel per twenty-four hours, which is equal to 0·47 per S.H.P. hour. For this speed there would be about 900 I.H.P. developed. The mean pressure with 90 revolutions

would be only 83·4 lbs. A sister ship, with modern good triple expansion engines, requires 12½ tons of Durham coal to do the same speed. If it cost 20 shillings per ton *f.o.b.*, the oil to equal it in cost would be at the rate of 71·4 shillings per ton, which is lower than was ruling for residuals. The great saving, however, in favour of the oil engine is in wages, etc., of firemen, which would amount to £20 to £25 per month in a low-power boat like the *Eavestone.*

On the other hand, it is pretty certain that engine-room accounts for overhauls and adjustments at terminal ports will be much heavier in the oil-engine ships; repairs and renewals will also figure largely. Moreover, the energies and attention of the engine-room staff will be more sorely tried, especially on voyages with lots of stoppages.

(3) *The Selandia* has twin-screw engines indicating in the aggregate 2500 I.H.P. at 140 revolutions per minute. Each engine has eight cylinders, 20·87 inches diameter × 28·74 inches stroke, four-cycle single-acting. The speed of this ship when partly loaded is 12 knots. At full speed the mean effective pressure in the cylinders is 90 lbs. This ship has made voyages to Java and back successfully, as have also her sister ships.

(4) *A Semi-Diesel engine*, having a cylinder 17 inches diameter × 27·5 inches stroke, when running at 180 revolutions per minute compressed to about 300 lbs. per square inch, and after explosion the pressure was 500 lbs. The thermal efficiency of this engine was found to be 40 per cent. and the mechanical 68 per cent., so that the general efficiency was 27·2 on the average throughout the trial.

(5) *A small Semi-Diesel* by Beardmore has four cylinders, 9 inches diameter × 13 inches stroke. It runs at 400 revolutions per minute, compresses to 150 lbs., and reaches 300 lbs. after explosion. A diagram taken from these engines when running on service shows a maximum pressure of 275 lbs. and a mean of 52·5, giving the ratio 5·24.

## MOTOR BOATS AND OIL ENGINES USING PETROL.

### Board of Trade Rules for.

### Instructions to Surveyors, 1913.

The following instructions are issued by the Board of Trade for the information and guidance of their officers in surveying motor boats for which a passenger certificate is required.

201. . . . The provisions of the Act as to passenger steamers, namely, section 267 and sections 271 to 288, apply, in so far as they are applicable, to motor boats, and therefore no motor boat may carry more than twelve passengers unless it has been surveyed within the preceding twelve months and holds a passenger certificate which is still in force, and is applicable to the voyage or excursion on which the vessel is about to proceed.

The Surveyor . . . must be satisfied that the hull and machinery are sufficient for the service intended and in good condition; and in considering this point, he must have regard to the conditions of wind and weather to which the vessel may be exposed, and must see that she is so constructed as to be able to meet these conditions without danger to human life.

. . . The Surveyor should be satisfied that proper precautions have been taken to guard against the occurrence of fire or explosion, and that adequate provision has been made for extinguishing fires should they occur.

202. **The duration of the certificate.** It is desirable in many cases to limit it to three months at first, but certificates for a longer period may be issued subsequently, if the engines are found to have worked satisfactorily, if the tank and pipe connections and fittings have proved oil-tight, and if everything in good condition.

203. The printed regulations relating to the survey of passenger steamers apply, as modified and supplemented by the provisions of this circular, sections 201–228.

The following requirements apply to launches or boats fitted with engines using petrol or other petroleum.

204. **Oil Tank.**—The oil tank should be well and substantially constructed of suitable material, and should not be larger than is reasonably necessary. When made of iron or steel, it should be efficiently galvanised externally; if it is proposed to galvanise it on the inside, the work should be carefully done, otherwise the zinc coating may become detached by the corrosive action of the petrol. The tank and its connections should be perfectly oil-tight, and should be tested by hydraulic pressure corresponding to a head of water of at least 15 feet.

Particulars of the tank and its fittings should be submitted, if a pressure feed system is employed.

205. **Tray for Oil Tank.**—The tank should be securely fixed in position, and should rest in a properly supported and suitable lead-lined or metal tray, situated above the deep-load line, with drain pipes from the tray leading overboard.

206. **Arrangements for filling Oil Tank.**—In order to reduce the risk of explosion, the arrangements for filling the tank should be such that oil will not readily be spilled, or overflow and drain into, or lodge in, either the compartment containing the tank or any other part of the vessel; and provision should be made whereby the petrol vapour which is displaced when the tank is replenished will be discharged overboard. If the tank is filled through the deck, the woodwork surrounding the inlet pipe should be covered with sheet metal to prevent it becoming saturated with oil or spirit. A properly secured wire-gauze diaphragm, or tube, which should be made readily removable for cleaning and examination, should be fitted to each inlet and outlet on the tank; and the filling pipe, or orifice, should be furnished with a suitable screwed cap.

No loose cans of petrol should be carried in the boat, and the permanent petrol tank should be charged with petrol when the passengers are not on board.

207. **Safety Device for Oil Tank.**—Means should be provided for relieving the pressure in the tank in case of fire. This may consist of (*a*) an open pipe with wire-gauze diaphragm fitted, (*b*) a light spring-loaded valve, or (*c*) a satisfactory fusible plug or similar safety device.

208. The pipe conveying the petrol to the carburettor should be solid drawn copper and provided with a flexible bend or bends; a cock or valve should be fitted at each end of the pipe. one on the tank, and the other on the carburettor or float chamber, and the joints and the couplings should be readily accessible and such that they can be made and kept perfectly oil-tight. In this connection it should be noted that coupling and other joints made of soft solder are not satisfactory.

The air inlet pipe to the carburettor should be fitted with a wire-gauze diaphragm and carried to the ship's side or to a vertical height of at least six feet above the carburettor, and to such a position that there will be no danger of ignition of any petrol vapour that may escape therefrom when the engine is stopped. It is desirable that carburettors should be of such a type that when the motor is stopped the supply of petrol to the carburettor will be shut off automatically. In some cases a suitable receptacle is required to be fitted to the carburettor to prevent an overflow of petrol from the latter into the launch when the engine is stopped; this receptacle should be formed with a narrow neck and provided with a wire-gauze covering at the mouth and with means for draining it.

The exhaust pipe should be efficiently cooled to prevent danger.

209. **Ignition.**—An exposed spark gap should not be permitted in the engine-room, and care should be taken that the leads from the accumulators or generators to the sparking plugs are efficiently insulated and well secured and protected from moisture, particularly when the high-tension system of electrical ignition is adopted.

Ignition tubes should not be passed unless oil having a higher flash point than 73° Fahrenheit is used. If blow lamps are used for this class of oil, they must be fixed and the flame enclosed.

210. **Motor Compartment, Ventilation, &c.**—If the motor, or petrol tank, is situated below deck, it should be confined within a separate watertight and well-ventilated compartment, in which no stove or other apparatus for containing fire should be placed. The compartment should be furnished with at least two cowl ventilators, so arranged as to prevent the accumulation of oil vapour in the lower part of the space, to which part one of the ventilators should extend. Any enclosed space within which the motor, or tank, is placed should be ventilated in this manner, except in the case of small open launches where louvres, or other suitable openings can be provided, in which case one cowl ventilator may be passed if the arrangements are to the Surveyor's satisfaction. In a vessel of this class, the space occupied

by the motor, petrol tank, &c., should, preferably, be at the after end of the boat, and separated from the space allotted for the accommodation of passengers and crew by a substantial bulkhead as high as the seats, and watertight for at least the lower half; but, if it is specially desired to place the motor amidships, or forward, either arrangement may be allowed, provided a bulkhead, formed in the manner stated, is placed between the motor space and the passenger or crew space.

211. **Tray for Motor.**—If the vessel is of wood, a metal tray which can readily be cleaned should be fitted under the motor; and, if flooring boards are provided, they should be closely fitted, but made removable in order to facilitate cleaning and inspection.

212. **Machinery to be fenced where necessary.**—The machinery should be fenced where necessary, in order to prevent injury to the persons in the boat.

213. The foregoing are the more important requirements relating to the construction and arrangement of oil motors, and those which must be complied with generally; but it should be clearly understood that the details of each case will be considered separately by the Board of Trade, and it is only when this has been done that all the requirements which must be fulfilled can be determined. It is therefore necessary that the Surveyors should be in a position to submit to the Board at an early stage full particulars of the case, including plans of the motor, oil-tank, carburettor, and reversing gear, in order that any necessary alterations may be pointed out before the construction of the machinery is commenced, if possible.

To ensure this being done, it is necessary that the owners or builders of motor boats or engines should make early application for survey, when such is required, by filling in the form Surveys 6 and sending it, with the necessary fee, to the nearest mercantile marine office.

In all cases the limits within which the vessel is intended to ply should be stated.

The material of which the various parts are made should be shown on the plans submitted. The steel portions should be tested in the presence of the Surveyor and should be of ductile quality, and the scantlings of all the parts should be sufficient for the purpose intended.

214. **The cylinders** should be satisfactorily tested by water pressure in the Surveyor's presence to double the maximum working pressure to which they will be subjected, and the silencer and exhaust pipes to at least one-fourth of the pressure applied to the cylinders, and they should, owing to the possibility of back-firing, be of ample strength.

215. **The boat should be tried** in the Surveyor's presence for handiness in manœuvring, going ahead, stopping and going astern, before the declaration is issued.

216. **The means for extinguishing fire** to be provided in the case of boats less than 30 feet in length should consist of at least one chemical fluid fire-extinguisher of an efficient size and pattern, and a box of sand of not less than one cubic foot in capacity; a suitable scoop for applying

the sand should also be provided. In larger boats, or in special circumstances, additional fire-extinguishing appliances may be required. The extinguishers should be of sufficient strength to withstand the pressure generated by the combination of the chemicals employed when the outlet valve to the vessel is closed. The extinguishing medium must be of a nature that will not be harmful to anyone with whom it comes in contact. Permanent instructions should be attached to each apparatus showing how it is operated, and it should be stated whether the apparatus may be stowed in any position or in a certain position only. The extinguishers should be placed in a position ready for immediate use, and the plunger for breaking the glass containing the acid should be protected from accidental movement.

It is desirable that petrol or other tanks containing highly volatile and inflammable liquids should be provided with a safety fitting to relieve the internal pressure, and prevent the tank from bursting in the event of its becoming overheated by fire. Whenever such fittings are about to be adopted, particulars should be first reported for consideration.

217. The foregoing instructions apply to all new motor boats and to all motor boats which come under survey for the first time. In the case of boats which have already been passed, the Surveyors need not insist on alterations being made in order to secure full compliance with the regulations, provided they are fully satisfied with the present arrangements ; but whenever any repairs or renewals are being effected, the regulations should be fully complied with. This does not apply to the requirements of paragraph 23 as to fire-extinguishing appliances, which must be complied with in all cases.

## LLOYD'S RULES FOR INTERNAL COMBUSTION MARINE ENGINES (other than Diesel Type).*

### General.

**Section 1.** In vessels propelled by internal combustion engines, the rules as regards machinery will be the same as those relating to steam engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

### Construction.

**Section 2.** 1. The following points should be observed in connection with the design of the engines.

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed and easily accessible for examination and adjustment.

4. In engines of above 60 B.H.P. which are not reversible and which are manœuvred by clutch, a governor or other arrangement must be fitted to prevent racing of the engine when declutched.

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* For Diesel type, Rules, &c., *vide* Appendix H, p. 695.

5. Efficient positive means of lubrication (preferably sight feed) must be fitted to each part requiring continuous lubrication.

6. If the engines are of the closed-in type, they must be so fitted that the contained lubricating oil can be drained when necessary, and in wood vessels an easily drained metal or metal-lined tray must be fitted to prevent leakage of either fuel oil or of lubricating oil from saturating the woodwork.

7. Carburettors, where petrol is used, and vaporisers, where paraffin is used, should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carburettor or vaporiser, a gauze-covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic gauze diaphragms should be fitted either between the carburettor (or vaporiser) and cylinders, or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and accumulator, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be enclosed; and the sparking coils must not be placed where they can be exposed to explosive vapours.

9. No exposed spark gap should be fitted.

10. In paraffin and heavy-oil engines where lamps are used for ignition or for vaporising, these lamps should be fixed by some suitable bracket, and the flame enclosed when in use.

11. The circulating pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position, and the circulating pipe is to be provided with an efficient strainer inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

12. The pumping arrangements are to be the same as would be required in the case of a steam vessel of the same size and power, with the exception that no bilge injection need be fitted. In the cases of vessels fitted with water ballast, the water ballast pump must have one direct suction from the engine-room bilges in addition. In open launches, and in small sailing vessels in which the engines are auxiliary only, a suitable additional hand pump, fitted to draw from the engine-room bilges, may be accepted in lieu of a power-driven pump.

13. The cylinders are to be tested by hydraulic pressure to twice the working pressure to which they will be subjected. The water jackets of the cylinders to 50 lbs. per square inch, and the exhaust pipes and silencer to 10 lbs. per square inch.

## Fuel Tanks and Connections.

**Section 4.** 1. Separate fuel tanks are to be tested with all fittings, to a head of at least 15 feet of water. If pressure-feed tanks are employed, they are to be tested to twice the working pressure which will come on them, but at least to a head of 15 feet of water. If the tanks are made of iron or steel they should be galvanised.

2. Strong and readily removable metallic gauze diaphragms should be fitted at all openings on petrol tanks.

3. Paraffin or heavy-oil tanks, not used under pressure, are to be fitted with air pipes leading above deck. Pressure-feed tanks and tanks containing petrol should be provided with escape valves discharging into pipes leading to the atmosphere above deck. The upper ends of all air pipes are to be turned down, and pipes above 1 inch diameter are to be provided with gauze diaphragms at the end.

4. No glass gauges are to be fitted to fuel tanks containing either petrol, paraffin, or heavy oil.

5. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

6. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For petrol tanks the trays must have drains leading overboard where possible, or they should be gauze-covered trays with means for draining them.

7. All fuel pipes are to be of annealed seamless copper with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carburettor or vaporiser. The fuel pipes should be led in positions where they are protected from mechanical injury and can be exposed to view throughout their whole length.

8. The engine-room, and the compartment in which the fuel tanks are situated are to be efficiently ventilated.

9. An approved fire-extinguishing apparatus must be supplied.

## SUPERHEATED STEAM.

Previous to the general adoption of the compound engine on shipboard and the raising of the working pressure to 60 lbs. per square inch, superheaters were in very general use both in the Navy and Mercantile Marine. The old box form of boiler, made for a pressure of 30 lbs., had fitted in the uptake some form of tubular receiver for steam which was exposed to the heat action of the effluent gases, so that its temperature was raised from 274° F. to as much as 400°; generally, however, it was not possible to get so high as this, and in the Mercantile Marine 360° was satisfactory. The economy due to this conversion of waste heat into energy was marked, but the life of the superheater was never very long, and the restrictions imposed by the Board of Trade latterly were onerous; shipowners were therefore content with the saving effected by the compound system and the high-pressure boiler, and so gave up the use of what could have very materially improved the efficiency of the new installations. To-day the temperature of the saturated steam generated for quadruple engines is 390° F., or as high as superheated steam was formerly, and the absolute pressure is nearly five times; but, notwithstanding, even superheating it is an advantage and can be effected at comparatively small cost. In fact, moderate superheat can be obtained still from the waste heat in the uptake; but to obtain the best economic results the grate area has to be such that the temperature in the uptake is considerably higher than is necessary for evaporative efficiency.

### Table LXXIIIa.—A Comparison of the Steam Consumption with and without Superheat when Working with Steam at 180 Lbs. Pressure in Lbs. per S.H.P. Hour.

| S.H.P. | Triple Compound Reciprocator. | | Ordinary Direct Drive Turbine. | | Combined Reciprocator and Turbine. | |
|---|---|---|---|---|---|---|
| | Vac. 26. Saturated Steam. | Vac. 26. Superheat to 572° F. | Vac. 28·5. Saturated Steam. | Vac. 28·5. Superheat to 572° F. | Vac. 28·5. Saturated Steam. | Vac. 28·5. Superheat to 572° F. |
| 500 | 16·50 | 11·75 | 14·00 | 11·75 | 18 to 22 | 13 to 15 |
| 1000 | 15·25 | 11·25 | 11·75 | 9·75 | 12 to 13 | 9·5 to 9·9 |
| 1500 | 14·75 | 11·00 | 11·25 | 9·50 | 11·25 | 9·00 |
| 2000 | 14·50 | 11·00 | 11·00 | 9·25 | 11·00 | 8·70 |

**The maximum temperature** for safe and efficient working in every-day practice is 550°, or about 160° F. above that of the saturated steam. Some engineers on shore use steam at 600° or even a little higher, but in doing so there is risk which is hardly worth running. Moreover, to attain so high a superheat there must be an independently fired superheater, and the gases on leaving it must be very hot.

**The specific heat of steam** is determined by this formula.

$$\text{Specific heat at constant pressure} = 0{\cdot}4304 + 0{\cdot}000378 \times T,$$

wherein T is the temperature of the steam in degrees Fahr.

### Table LXXIV.—Specific Heat of Steam at various Temperatures.

| Temp. | Specific Heat. | Temp. | Specific Heat. | Temp. | Specific Heat. | Temp. | Specific Heat. | Temp. | Specific Heat. |
|---|---|---|---|---|---|---|---|---|---|
| F.° | | F.° | | F.° | | F.° | | F°. | |
| 300 | 0·5438 | 380 | 0·5740 | 460 | 0·6043 | 540 | 0·6345 | 620 | 0·6648 |
| 310 | 0·5476 | 390 | 0·5778 | 470 | 0·6081 | 550 | 0·6383 | 630 | 0·6685 |
| 320 | 0·5514 | 400 | 0·5816 | 480 | 0·6118 | 560 | 0·6421 | 640 | 0·6723 |
| 330 | 0·5551 | 410 | 0·5854 | 490 | 0·6156 | 570 | 0·6459 | 650 | 0·6761 |
| 340 | 0·5589 | 420 | 0·5894 | 500 | 0·6194 | 580 | 0·6496 | 660 | 0·6799 |
| 350 | 0·5627 | 430 | 0·5929 | 510 | 0·6232 | 590 | 0·6534 | 670 | 0·6837 |
| 360 | 0·5665 | 440 | 0·5967 | 520 | 0·6270 | 600 | 0·6572 | 680 | 0·6874 |
| 370 | 0·5703 | 450 | 0·6005 | 530 | 0·6307 | 610 | 0·6610 | 690 | 0·6912 |

**The amount of heat required to superheat** a pound of steam to a temperature $T_s$ from its natural temperature T may be ascertained by taking the specific heat at the mean temperature and multiplying by the difference in temperature. Thus :—

$$(a) \qquad \text{Mean temperature} = \frac{T + T_s}{2}.$$

$$(b) \text{ Amount of heat to raise temperature} = \text{sp. heat of } \frac{T + T_s}{2} (T_s - T).$$

*Example.*—To superheat a pound of steam at a pressure of 180 lbs., or 195 absolute, to 500° F.

$$\text{Mean temperature} = \frac{380^\circ + 500^\circ}{2} = 440^\circ.$$
$$\text{Specific heat } 440^\circ = 0{\cdot}5967.$$
$$\text{Difference in temperature} = 500^\circ - 390^\circ = 110^\circ.$$
$$\text{Heat required} = 0{\cdot}5967 \times 110^\circ \text{ or } 65{\cdot}637 \text{ B.T.U.}$$

**The total heat of superheated steam** may be calculated by Prof. Peabody's formula, which is as follows,—

$$\text{Total heat} = 0{\cdot}4805\ (T_a - 10{\cdot}38\sqrt[4]{P}) + 857.$$

$T_a$ is the absolute temperature Fahr.; P is the absolute pressure.

Or by Prof. Ripper's formula,

$$\text{Total heat} = H + 0{\cdot}48\ (t_s - t_1),$$

where H is the total heat of saturated steam at a pressure $p$ and a temperature $t_1$, and $t_s$ is the temperature to which it is raised by superheating.

*Example.*—The total heat of steam at 200 lbs. gauge pressure and 500° F. temperature will be

$$H = 0{\cdot}4805\left\{(500 + 461) - 10{\cdot}38\ (200 + 15)^{\frac{1}{4}}\right\} + 857.$$
$$= 442{\cdot}7 + 857 = 1299{\cdot}7^\circ \text{ F.}$$

**Superheated steam transmitted** through pipes is greater than is the case with saturated steam of same pressure. The quantity may be estimated by the following formula.

$d$ is the diameter of the pipe in inches, whose length is L feet; W is the weight of a cubic foot of the steam at a pressure $P_1$ at entry and $P_2$ at exit

$$\text{Weight passed per minute} = 87\sqrt{\frac{W(P_1 - P_2)d^5}{L\left(1 + \frac{3{\cdot}6}{d}\right)}}.$$

## Table LXXV.—Maximum Work possible from 1 lb. of Superheated Steam expanding to and exhausting at 1 lb. pressure B.T.U.

| Initial Pressure Absolute. | Temperature of Steam after superheat, degrees Fahr. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 350 | 375 | 400 | 425 | 450 | 475 | 500 | 525 | 550 | 575 | 600 | 625 | 650 | 675 |
| Lbs. | | | | | | | | | | | | | | |
| 100 | 300·0 | 315·9 | 329·0 | 344·0 | 359·0 | 374·0 | 389·5 | 405·0 | 421·0 | 437·0 | 453·0 | 470·0 | 486·5 | 503·0 |
| 125 | 306·5 | 321·5 | 335·0 | 349·5 | 364·0 | 379·0 | 394·5 | 410·0 | 426·0 | 442·0 | 458·0 | 475·0 | 491·3 | 508·0 |
| 150 | .. | 326·0 | 339·0 | 353·0 | 368·0 | 383·0 | 398·0 | 414·0 | 430·0 | 446·0 | 462·0 | 478·5 | 495·0 | 512·0 |
| 175 | .. | 329·3 | 342·0 | 356·5 | 375·0 | 386·5 | 401·0 | 417·3 | 433·0 | 449·3 | 465·3 | 482·0 | 498·5 | 515·5 |
| 200 | .. | 332·0 | 344·5 | 358·5 | 373·5 | 389·0 | 404·0 | 420·0 | 436·0 | 452·0 | 468·0 | 484·0 | 501·0 | 518·0 |
| 225 | .. | .. | 346·4 | 360·5 | 355·5 | 391·0 | 406·0 | 422·0 | 438·0 | 454·0 | 470·0 | 486·3 | 502·7 | 520·0 |
| 250 | .. | .. | 348·0 | 362·0 | 377·5 | 392·5 | 408·9 | 423·5 | 439·5 | 455·0 | 471·5 | 488·0 | 504·0 | 521·0 |
| 275 | .. | .. | .. | 363·0 | 379·0 | 394·0 | 409·5 | 425·0 | 441·0 | 456·4 | 472·8 | 489·3 | 505·0 | 522·0 |
| 300 | .. | .. | .. | 364·0 | 380·0 | 395·0 | 410·5 | 426·0 | 442·0 | 457·1 | 473·4 | 490·0 | 505·6 | 522·6 |

## Table LXXVa.—Weight of Superheated Steam passed per minute in lbs. through Pipes 240 diameters long, with a drop of 1 lb. Pressure.

| Gauge Pressure. | Diameter of Pipe in Inches. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 |
| Lbs. | | | | | | | | | | |
| 100 | 5·25 | 26·0 | 64·2 | 118·5 | 196 | 293 | 400 | 535 | 690 | 863 |
| 120 | 5·63 | 27·9 | 69·9 | 127·2 | 210 | 315 | 433 | 574 | 740 | 925 |
| 140 | 6·00 | 29·7 | 73·2 | 135·0 | 223 | 334 | 458 | 609 | 786 | 980 |
| 160 | 6·31 | 31·3 | 77·4 | 142·0 | 235 | 352 | 483 | 640 | 825 | 1033 |
| 180 | 6·64 | 33·0 | 81·5 | 148·9 | 245 | 368 | 503 | 670 | 865 | 1080 |
| 200 | 6·95 | 34·5 | 85·7 | 155·8 | 256 | 383 | 525 | 698 | 903 | 1125 |
| 220 | 7·25 | 36·0 | 89·8 | 162·3 | 266 | 398 | 545 | 726 | 940 | 1172 |
| 240 | 7·55 | 37·5 | 93·8 | 168·4 | 275 | 412 | 563 | 752 | 983 | 1214 |

**The heating surface of a superheater** per pound of steam must depend on the difference in temperature between the hot gases and the steam.

With considerable difference 6 B.T.U.'s are transferred per square foot of surface per hour for each degree of difference Fahr., when the surfaces within and without are clean.

That is, if steam of 200 lbs. pressure, 500° temperature, is passing through a superheater exposed to hot gas at 650° temperature,

Each square foot should pass

$$6\,(650-500) \text{ or } 900 \text{ B.T.U. per hour.}$$

Each pound of steam requires $(500-387)\ 0{\cdot}599=67{\cdot}7$ B.T.U., so that if the consumption of steam is 12 lbs. per I.H.P., the surface per I.H.P. $=67{\cdot}7\times12\div900$, or $0{\cdot}9027$ square feet.

## SKIN FITTINGS AND VALVES.

**Blow-off valves.**—These should always be bronze stop-valves opening inwards, and should have spigots passing through the ship's plating. If any guard cock is fitted, it should be placed between the valve on the boiler and the sea-valve, and independent of both.

The method of attachment to the ship's skin, recommended by Lloyd's, is shown in figs. 43, and 44 ; the Admiralty method in figs. 45 and 46.

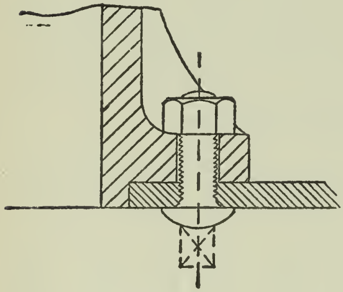

Fig. 43.

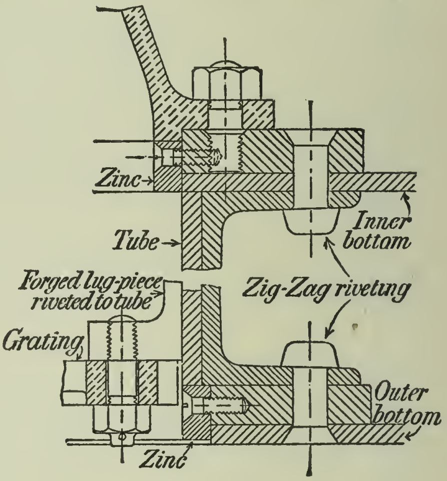

Fig. 44.

Where the valve has to open a communication through a double bottom, as is usually the case in Naval work, one of the Kingston type is generally employed, attached to the outer bottom, as shown in fig. 46, and made tight by means of a stuffing box, where it passes through the inner bottom.

The use of cast-iron for boxes or casings of blow-off valves is forbidden by the Board of Trade rules.

**Discharge valves.**—Here also, with the object of reducing the number of openings in the ship's skin, the smaller valves should, where possible, discharge into the casings of the larger ones, the outlets from the latter being increased in area as necessary.

In the Merchant Service, lifting non-return valves with cast-iron casings are generally used ; but in the Navy, though the same type of valve is used, the only material permitted is bronze.

Provision should be made for hanging up the valve, but, when the circulating pumps are of the reciprocating type, it should not be possible to hold it on its seat in any way.

All discharge valves should have spindles passing through the covers, and cross handles for turning them round on their seats.

In some cases, where room is pinched, a swinging flap-valve may be employed with advantage; and where the pumps are of the centrifugal type, either valves of this class or sluice valves should be used, in order to reduce the resistances as much as possible.

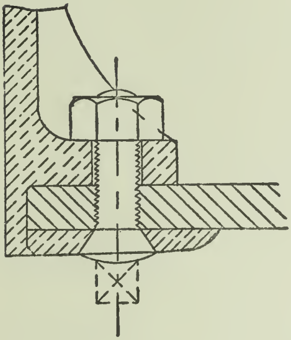

Fig. 45.

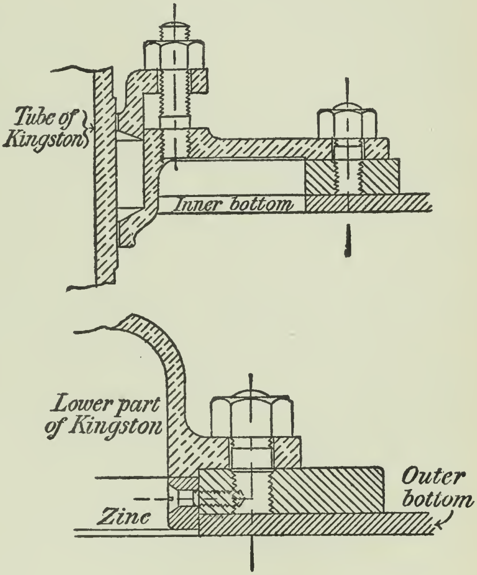

Fig. 46.

There should always be a spigot passing through the skin plating of the vessel, and, where the valve is large, it is often necessary to have a compensating plate inside riveted to the skin plate.

**The Board of Trade regulations** referring to sea connections are as follows :—

194. All inlets or outlets in the bottom or side of a vessel near to, at, or below the deep-load water line other than the outlets of water-closet, soil, scupper, lavatory, and urinal pipes should have cocks or valves fitted between the pipes and the ship's side or bottom; such cocks or valves should be attached to the skin of the ship, and be so arranged that they can be easily and expeditiously opened or closed at any time; and the cocks, valves, and the whole length of the pipes should be accessible at all times.

Cocks or valves standing exceptional distances from the ship's plating, that is where the necks are longer than is necessary for making the joint, should not be passed without the sanction of the Board of Trade, and one condition of their being passed is that they should be made of gun-metal, and well bracketed.

188. With a view to the prevention of accidents to boilers through the blow-off cocks being left open after the boiler is run up, and to prevent water getting accidentally into the ship by cocks being left open, all blow-off cocks and sea connections below the plates or out of sight should be fitted with a guard over the plug, with a featherway in the same, and a key on the spanner, so that the spanner cannot be removed unless the cock is closed. The spanner, &c., when in place should extend to the platform. When cocks are in sight, guards need not be fitted provided the spanners are secured to the plugs by pins. The spanners should not be shrunk on the heads of the plugs. One cock should be fitted to the boiler, and another cock on the skin of the ship or on the side of the Kingston valve. Valves may be substituted for the blow-off cocks on the boilers, but, in such cases, the blow-off cock on the skin of the ship must be fitted with a spanner guard so that the spanner cannot be removed when the cock is open; and, if the blow-off pipe is used for more than one boiler, an intermediate switch-cock, or suitable non-return valve to each boiler, should be fitted so that water cannot be blown from one boiler to another.

189. When the pipes are so arranged that the water in the boilers can be circulated by means of the donkey pump, similar precautions should be observed; and a cock fitted with spanner and guard, the handle of which will stand above the level of the platform, should be fitted in the circulating pipes, preferably near the pump.

190. A non-return valve having a screw spindle not attached, by which the valve may be set down on its seat when necessary, should be fitted to all pipes which are so led or placed that water could, unless such non-return valves were fitted, run from the boiler or the sea into the bilge, either by accidentally or intentionally leaving a cock or valve open; the only exception to this requirement has reference to the fireman's ash cock, which must have a cock or valve on the ship's side and be above the stoke-hold plates.

191. Cast-iron stand-pipes or cocks intended for the passage through them of hot brine should not be passed.

Surveyors should also discourage the use of cast-iron chocks and saddles for boilers, and particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

192. The exhaust pipe for the donkey engine should not be led through the ship's side, but should be led on deck or into the main waste steam pipe,* and in all cases should have a drain-cock on it.

193. In the case of the outlets of water-closet, soil, scupper, lavatory and urinal pipes which are below the weather deck, there should be an

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* AUTHORS' NOTE.—This plan renders overhaul of main safety-valves difficult when vessel is in port; a separate waste pipe should always be fitted.

elbow of good substantial metal other than cast-iron or lead ; and the pipe connected with this elbow should, if of lead, have a sufficient bend to provide for expansion in the pipe or any movement from the working of the ship. Pipes, no matter of what material they may be constructed, should never be fitted in a direct line between the apertures in the ship's side and their connection with the deck, or closet, or other fittings. The pipes and valves should be protected from the cargo by a substantial casing of wood or iron, which need not be watertight.

**Lloyd's Rules** relating to sea connections are as follows :—

5. Cocks and valves connecting all suction pipes to be fixed above the stokehold and engine-room platforms.

6. The arrangements of pumps, bilge injections, suction and delivery pipes, is to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect.

*Sec.* 7.—8. All discharge pipes to be, if possible, carried above the deep-load line, and to have discharge valves fitted on the plating of the vessel in an accessible position.

*Sec.* 7.—9. No pipes to be carried through the bunkers without being properly protected.

*Sec.* 9.—2. All sea-cocks to be fitted on the plating of the vessel above the level of the stoke-hole and engine-room platforms, or attached to Kingston valves of a height sufficient to lift them up to the level of these platforms.

36. The bolts securing all cocks or sea connections to the plating of the vessel are to be tapped into the plating of the vessel or fitted with countersunk heads.

37. The blow-off cocks on the plating of the vessel are to be fitted with spigots passing through the plating, and a brass or gun-metal ring on the outside. The cocks are to be so constructed that the key or spanner can only be taken off when the cock is shut.

See also paragraph 6 of section 27 of Lloyd's rules.

## Table LXXVI.—Results of Trials of Triple-Expansion Three-Crank Engines.

| Ship. | Number of Screws. | Cylinders. | | | | Engine Room. | | | | | | | | One Engine Only. | | | | Nominal Rate of Expansion. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diameters. | | | Stroke. | Steam. | Vacuum. | Revolutions. | Cut-off in H.P. Cylinders. | Mean Pressures. | | | | Indicated Horse-power. | | | | |
| | | H.P. | M.P. | L.P. | | | | | | H.P. | M.P. | L.P. | Referred to L.P. Cylr. | H.P. | M.P. | L.P. | Total. | |
| | | ins. | ins. | ins. | ins. | lbs. | ins. | | | lbs. | lbs. | lbs. | lbs. | | | | | |
| S.S. Ag. V. . | Twin | 41 | 67 | 106 | 63 | 149 | 24·0 | 78·0 | 0·78 | 62·6 | 27·8 | 11·16 | 32·8 | 2214 | 2470 | 2494 | 7178 | 8·58 |
| S.S. Krsn. . | ,, | 36 | 57 | 92 | 54 | 155 | 27·3 | 90·6 | .. | 82·9 | 36·8 | 14·2 | 41·0 | 2085 | 2320 | 2335 | 6740 | .. |
| S.S. Lgra. . | ,, | 33 | 54 | 88 | 60 | 183 | 25 | 72·0 | .. | 77·5 | 30·0 | 12·0 | 34·2 | 1444 | 1490 | 1592 | 4535 | .. |
| H.M.S. Ldn. . | ,, | 31½ | 51½ | 84 | 51 | 245 | 25·5 | 110 | .. | 115·0 | 44·0 | 17·9 | 50·0 | 2543 | 2601 | 2815 | 7859 | .. |
| H.M.S. Argt. . | ,, | 26 | 42 | 68 | 39 | 245 | 26·0 | 143 | 0·70 | 112·5 | 47·1 | 20·5 | 52·8 | 1685 | 1841 | 2100 | 5406 | 9·80 |
| S.S. Czgo. . | ,, | 32 | 48 | 80 | 48 | 145 | 26·0 | 71·0 | 0·66 | 68·1 | 32·0 | 11·0 | 33·6 | 943 | 994 | 947 | 2884 | 9·50 |
| S.S. Otlo. . . | ,, | 24¾ | 40 | 70 | 48 | 200 | 28·0 | 71·0 | 0·59 | 88·5 | 36·8 | 13·6 | 36·7 | 733 | 794 | 903 | 2430 | 15·25 |
| S.S. R.W. . | Single | 25 | 40 | 68 | 45 | 180 | 27·5 | 61·0 | .. | 68·7 | 27·1 | 10·15 | 30·6 | 500 | 502 | 545 | 1547 | .. |
| S.S. Vna. . . | Twin | 26 | 39½ | 61 | 36 | 155 | 26·5 | 135 | 0·72 | 62·4 | 31·2 | 14·15 | 38·0 | 810 | 933 | 1011 | 2754 | 7·80 |
| S.S. Ped. . . | ,, | 19 | 30 | 48 | 24 | 180 | 25·0 | 192 | 0·68 | 81·0 | 33·5 | 13·7 | 39·6 | 536 | 554 | 577 | 1667 | 9·37 |
| S.S. Cstr. . . | Single | 19 | 33 | 56 | 36 | 200 | 25·0 | 91·5 | 0·59 | 87·8 | 33·5 | 10·9 | 32·5 | 417 | 470 | 445 | 1333 | 14·70 |
| H.M.S. Snpr. . | Twin | 19¼ | 28½ | 43 | 18 | 180 | 25·0 | 371 | 0·58 | 64·5 | 28·8 | 14·6 | 39·5 | 635 | 592 | 715 | 1942 | 8·62 |
| H.M.S. Chgr. . | ,, | 18 | 26 | 39½ | 18 | 160 | 25·0 | 363 | 0·81 | 62·0 | 37·8 | 13·5 | 41·9 | 523 | 626 | 547 | 1696 | 6·00 |
| S.S. Mlpo. . | Single | 13 | 22 | 36 | 24 | 200 | 23·5 | 112 | 0·60 | 96·7 | 38·7 | 13·12 | 40·0 | 174 | 199 | 181 | 554 | 12·75 |
| S.S. Iona . . | ,, | 22 | 34 | 57 | 39 | 150 | 27·8 | 61·5 | 0·25 | 47·0 | 20·0 | 7·2 | 21·7 | 208 | 217 | 220 | 645 | 26·88 |
| S.S. Artc. . | ,, | 12 | 18 | 30 | 18 | 155 | 28·0 | 136 | 0·62 | 67·8 | 29·1 | 11·4 | 32·5 | 92·1 | 91·4 | 99·5 | 283 | 10·1 |
| S.S. Zde. . . | Twin | 12½ | 19 | 32 | 24 | 175 | 27·0 | 148 | 0·68 | 69·0 | 36·5 | 13·75 | 36·8 | 152 | 182 | 197 | 531 | 9·62 |
| S.S. Gte. . . | Single | 12 | 20 | 32 | 23 | 200 | 22·0 | 136 | 0·58 | 74·5 | 38·8 | 15·50 | 41·0 | 133 | 192 | 197 | 522 | 12·26 |

Table LXXVI. (continued).—Results of Trials of Triple-Expansion Four-Crank Engines.

| Ship. | Number of Screws. | Cylinders. | | | | | Engine Room. | | | | | | | | | One Engine only. | | | | | Nominal Rate of Expansion. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diameter. | | | | Stroke. | Steam. | Vacuum. | Revolutions. | Cut-off in H.P Cylinders. | Mean Pressures. | | | | | Indicated Horse-power. | | | | | |
| | | H.P. | M.P. | L.P. | L.P. | | | | | | H.P. | M.P. | L.P. | L.P. | Ref. L.P. | H.P. | M.P. | L.P. | L.P. | Total. | |
| | | ins. | ins. | ins. | ins. | ins. | lbs. | ins. | | | lbs. | lbs. | lbs. | lbs. | lbs. | | | | | | |
| H.M.S. Gh. . | Twin | 43½ | 71 | 81½ | 81½ | 48 | 278 | 27·0 | 126 | .. | 105 | 44·6 | 16·9 | 17·9 | 50·8 | 4764 | 5399 | 2698 | 2849 | 15,710 | .. |
| H.M.S. Wr. | ,, | 43½ | 69 | 77 | 77 | 42 | 194 | 26·7 | 138 | .. | 72·6 | 32·7 | 18·5 | 19·2 | 44·1 | 3147 | 3568 | 2513 | 2607 | 11,834 | .. |
| H.M.S. Am. | ,, | 41½ | 65½ | 73½ | 73½ | 42 | 185 | 25·0 | 143 | .. | 82·6 | 32·5 | 16·7 | 16·6 | .. | 3383 | 3315 | 2143 | 2136 | 10,977 | .. |
| H.M.S. Cd. . | ,, | 37 | 60 | 69 | 69 | 42 | 296 | 26·5 | 145 | .. | 105·2 | 41·3 | 19·1 | 18·8 | 49·3 | 3484 | 3594 | 2194 | 2170 | 11,442 | .. |
| H.M.S. Dm. | ,, | 34 | 55½ | 64 | 64 | 48 | 245 | 27·0 | 120 | .. | 90·0 | 34·0 | 21·9 | 21·7 | 50·3 | 2370 | 2386 | 2044 | 2025 | 8,825 | .. |
| H.M.S. Fd. . | ,, | 31½ | 50½ | 57 | 57 | 30 | 227 | 25·3 | 212 | .. | 109 | 34·5 | 14·9 | 14·8 | .. | 2727 | 2224 | 1218 | 1217 | 7,386 | .. |
| H.M.S. Tz. . | ,, | 24¼ | 38½ | 42½ | 42½ | 24 | 271 | 24·0 | 246 | .. | 111 | 47·2 | 21·5 | 22·7 | 57·3 | 1534 | 1637 | 897 | 946 | 5,014 | .. |
| S.S. Par. . | ,, | 26 | 40 | 45 | 45 | 30 | 177 | 27·0 | 177 | 0·69 | 57·5 | 30·3 | 15·4 | 15·3 | 36·9 | 818 | 1019 | 682 | 630 | 3,149 | 8·89 |
| S.S. Cr. . | ,, | 14½ | 23 | 27 | 27 | 33 | 180 | 26·5 | 145 | 0·75 | 83 | 38·0 | 18·4 | 18·2 | 44·6 | 331 | 382 | 255 | 252 | 1,220 | 9·33 |
| H.M.S. Dv. . | ,, | 19½ | 30 | 34 | 34 | 18 | 206 | 25·0 | 385 | .. | 77·3 | 45·5 | 17·7 | 17·2 | .. | 808 | 1127 | 563 | 547 | 3,045 | .. |
| H.M.S. Sr. . | ,, | 18¾ | 27¼ | 28 | 28 | 18 | 194 | 24·1 | 391 | .. | 63·7 | 33·0 | 19·0 | 18·7 | .. | 626 | 684 | 412 | 406 | 2,128 | .. |

## Table LXXVI. (continued).—Results of Trials of Quadruple-Expansion Engines.

| Ship. | Number of Screws. | Cylinders. | | | | | Engine Room. | | | | | | | | | One Engine only. | | | | | Nominal Rate of Expansion. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diameters. | | | | Stroke. | Steam. | Vacuum. | Revolutions. | Cut-off in H.P. Cylinders. | Mean Pressures. | | | | | Indicated Horse-power. | | | | | |
| | | H.P. | M.P$_1$ | M.P$_2$ | L.P. | | | | | | H.P. | M.P$_1$ | M.P$_2$ | L.P. | Refd to L.P. | H.P. | M.P$_1$ | M.P$_2$ | L.P. | Total. | |
| | | ins. | ins. | ins. | ins. | ins. | lbs. | ins. | | | lbs. | lbs. | lbs. | lbs. | lbs. | | | | | | |
| S.S. Sxa. . | Twin | 29 | 41½ | 59 | 84 | 54 | 192 | 25·3 | 78 | .. | 73·1 | 41·0 | 20·9 | 12·08 | 39·0 | 985 | 1158 | 1202 | 1420 | 4765 | .. |
| S.S. Ptn. . | Single | 24 | 34½ | 49 | 71 | 48 | 220 | 28·0 | 65 | .. | 80·3 | 32·1 | 16·0 | 9·12 | 33·5 | 572 | 476 | 472 | 568 | 2088 | .. |
| S.S. Vsr. . | ,, | 26½ | 37½ | 59 | 77 | 57 | 230 | 27·5 | 75 | .. | 80·6 | 44·7 | 18·4 | 10·6 | 41·3 | 964 | 1068 | 1059 | 1068 | 4159 | .. |
| S.S. Oho. . | ,, | 22½ | 32 | 45 | 64 | 42 | 200 | 27·5 | 67·5 | 0·64 | 63·5 | 28·5 | 20·4 | 11·0 | 39·0 | 362 | 328 | 464 | 507 | 1661 | 12·66 |
| S.S. Rgi. . | ,, | 17 | 25 | 35 | 50 | 27 | 184 | 25·0 | 72·0 | 0·57 | 61·3 | 32·6 | 20·1 | 7·8 | 32·7 | 136 | 157 | 181 | 151 | 633 | 15·18 |
| S.S. Plo. . | ,, | 12¾ | 18½ | 27 | 39 | 24 | 200 | 25·0 | 98 | 0·62 | 75·0 | 34·3 | 15·3 | 11·85 | 34·8 | 113 | 110 | 104 | 168 | 496 | 15·05 |
| S.S. Cyk. . | ,, | 20 | 28 | 40 | 57 | 42 | 160 | 24·8 | 48·5 | 0·60 | 46·1 | 26·9 | 14·3 | 6·98 | 26·1 | 149 | 169 | 183 | 181 | 682 | 13·5 |
| S.S. Tk. . | ,, | 20 | 28 | 39½ | 57 | 33 | 212 | 25·0 | 140 | 0·63 | 73·5 | 37·0 | 19·8 | 10·0 | 40·0 | 540 | 532 | 565 | 596 | 2233 | 12·9 |
| S.S. Hl. . | ,, | 22½ | 31½ | 45 | 66 | 51 | 215 | 25·5 | 76 | 0·57 | 74 | 35·5 | 18·3 | 8·3 | 33·3 | 570 | 542 | 566 | 550 | 2228 | 15·1 |

# LLOYD'S REGISTER RULES FOR COPPER, IRON, AND STEEL STEAM PIPES.

In all steam pipes provision is to be made for expansion and contraction to take place without unduly straining the pipes. It is recommended that the various lengths of pipes should be made short of the designed lengths by amounts equal to half the calculated expansion by the temperature of the steam.

Copper steam pipes are to be tested by hydraulic pressure to twice the working pressure, and electro-deposited copper pipes are not to be approved for steam pipes.

Wrought iron or steel lapwelded pipes and seamless steel pipes are to be tested by hydraulic pressure to three times the working pressure. The flanges are to be forged out of the solid, and are to be fitted to the pipes either by being screwed with a vanishing thread, riveted, welded or expanded by an approved process.

The thickness of these pipes is not to be less than that given by the following formula:—

$$\text{Working pressure in lbs. per sq. inch} = C\,\frac{(T - \cdot 125)}{D}$$

where T = thickness of pipes in inches,
D = internal diameter of pipes in inches,

$$C \begin{cases} = 9{,}000 \text{ for lapwelded pipes,} \\ = 12{,}000 \text{ for seamless pipes.} \end{cases}$$

$$\text{Hydraulic test pressure of copper pipes in lbs. per sq. inch} = 4500\,\frac{T}{D}.$$

Rules recommended by the B.M.E.D. & C. Committee for pipes made of arsenious or other equally strong copper.

Working pressure $= (t - 3) \times F \div D$.

D is the internal diameter in inches, $t$ the thickness in 100th of an inch. F, for solid-drawn steam pipes, 60 ; for brazed, 48 ; for feed pipes, F is 48 and 40 respectively. No pipes to be less than $3\sqrt{D} + 1$.

Solid cold-drawn steel pipes, working pressure $= (t - 8) \times 120 \div D$.
,, hot- ,, ,, ,, $= (t - 12) \times 120 \div D$.
Welded iron or ,, ,, $= (t - 10) \times 90 \div D$.

No iron or steel pipes to be less than $5\sqrt{D} + 2$. Feed pipes 20 per cent. thicker than given by above rates.

## Table LXXVII.—Wire Gauges and their Equivalents.

| L. S. G. Legal Standard Wire Gauge. | | B. W. G. Birmingham Wire Gauge. | | W. W. G. Whitworth Wire Gauge. | |
|---|---|---|---|---|---|
| No. of Gauge. | Equivalent in ins. | No of Gauge. | Equivalent in ins. | No. of Gauge. | Equivalent in ins. |
| 7/0 | ·500 | 5/0 | ·500 | 300 | ·300 |
| 6/0 | ·464 | 4/0 | ·460 | 280 | ·280 |
| 5/0 | ·432 | 3/0 | ·420 | 260 | ·260 |
| 4/0 | ·400 | 2/0 | ·380 | 250 | ·250 |
| 3/0 | ·372 | 1/0 | ·340 | 240 | ·240 |
| 2/0 | ·348 | 1 | ·300 | 220 | ·220 |
| 0 | ·324 | 2 | ·280 | 200 | ·200 |
| 1 | ·300 | 3 | ·260 | 180 | ·180 |
| 2 | ·276 | 4 | ·240 | 165 | ·165 |
| 3 | ·252 | 5 | ·220 | 150 | ·150 |
| 4 | ·232 | 6 | ·200 | 135 | ·135 |
| 5 | ·212 | 7 | ·180 | 125 | ·125 |
| 6 | ·192 | 8 | ·164 | 120 | ·120 |
| 7 | ·176 | 9 | ·148 | 110 | ·110 |
| 8 | ·160 | 10 | ·132 | 100 | ·100 |
| 9 | ·144 | 11 | ·120 | 95 | ·095 |
| 10 | ·128 | 12 | ·108 | 90 | ·090 |
| 11 | ·116 | 13 | ·096 | 85 | ·085 |
| 12 | ·104 | 14 | ·084 | 80 | ·080 |
| 13 | ·092 | 15 | ·072 | 75 | ·075 |
| 14 | ·080 | 16 | ·064 | 70 | ·070 |
| 15 | ·072 | 17 | ·056 | 65 | ·065 |
| 16 | ·064 | 18 | ·048 | 60 | ·060 |
| 17 | ·056 | 19 | ·040 | 55 | ·055 |
| 18 | ·048 | 20 | ·036 | 50 | ·050 |
| 19 | ·040 | 21 | ·032 | 45 | ·045 |
| 20 | ·036 | 22 | ·028 | 40 | ·040 |
| 21 | ·032 | 23 | ·024 | 38 | ·038 |
| 22 | ·028 | 24 | ·022 | 36 | ·036 |
| 23 | ·024 | 25 | ·020 | 34 | ·034 |
| 24 | ·022 | 26 | ·018 | 32 | ·032 |
| 25 | ·020 | 27 | ·016 | 30 | ·030 |
| 26 | ·018 | 28 | ·014 | 28 | ·028 |
| 27 | ·0164 | 29 | ·013 | 26 | ·026 |
| 28 | ·0148 | 30 | ·012 | 24 | ·024 |
| 29 | ·0136 | 31 | ·011 | 22 | ·022 |
| 30 | ·0124 | 32 | ·010 | 20 | ·020 |
| 31 | ·0116 | 33 | ·009 | 19 | ·019 |
| 32 | ·0108 | 34 | ·008 | 18 | ·018 |
| 33 | ·0100 | 35 | ·007 | 17 | ·017 |
| 34 | ·0092 | 36 | ·006 | 16 | ·016 |
| 35 | ·0084 | 37 | ·005 | 15 | ·015 |
| 36 | ·0076 | 38 | ·004 | 14 | ·014 |
| 37 | ·0068 | 39 | ·003 | 13 | ·013 |
| 38 | ·0060 | 40 | ·002 | 12 | ·012 |

The following Table shows the Legal Standard Wire Gauge and its equivalents in millimetres :—

### Table LXXVIII.—Legal Standard Wire Gauge and Metric Equivalents.

| Number of Gauge. | Equivalents. | | Number of Gauge. | Equivalents. | |
|---|---|---|---|---|---|
| | Inch. | Millimetre. | | Inch. | Millimetre. |
| 7/0 | ·500 | 12·700 | 23 | ·024 | ·610 |
| 6/0 | ·464 | 11·785 | 24 | ·022 | ·559 |
| 5/0 | ·432 | 10·973 | 25 | ·020 | ·508 |
| 4/0 | ·400 | 10·160 | 26 | ·018 | ·457 |
| 3/0 | ·372 | 9·449 | 27 | ·0164 | ·4166 |
| 2/0 | ·348 | 8·839 | 28 | ·0148 | ·3759 |
| 0 | ·324 | 8·229 | 29 | ·0136 | ·3454 |
| 1 | ·300 | 7·620 | 30 | ·0124 | ·3150 |
| 2 | ·276 | 7·010 | 31 | ·0116 | ·2946 |
| 3 | ·252 | 6·401 | 32 | ·0108 | ·2743 |
| 4 | ·232 | 5·893 | 33 | ·0100 | ·2540 |
| 5 | ·212 | 5·385 | 34 | ·0092 | ·2337 |
| 6 | ·192 | 4·877 | 35 | ·0084 | ·2134 |
| 7 | ·176 | 4·470 | 36 | ·0076 | ·1930 |
| 8 | ·160 | 4·064 | 37 | ·0068 | ·1727 |
| 9 | ·144 | 3·658 | 38 | ·0060 | ·1524 |
| 10 | ·128 | 3·251 | 39 | ·0052 | ·1321 |
| 11 | ·116 | 2·946 | 40 | ·0048 | ·1219 |
| 12 | ·104 | 2·642 | 41 | ·0044 | ·1118 |
| 13 | ·092 | 2·337 | 42 | ·0040 | ·1016 |
| 14 | ·080 | 2·032 | 43 | ·0036 | ·0914 |
| 15 | ·072 | 1·829 | 44 | ·0032 | ·0813 |
| 16 | ·064 | 1·626 | 45 | ·0028 | ·0711 |
| 17 | ·056 | 1·422 | 46 | ·0024 | ·0610 |
| 18 | ·048 | 1·219 | 47 | ·0020 | ·0508 |
| 19 | ·040 | 1·016 | 48 | ·0016 | ·0406 |
| 20 | ·036 | ·914 | 49 | ·0012 | ·0305 |
| 21 | ·032 | ·813 | 50 | ·0010 | ·0254 |
| 22 | ·028 | ·711 | ... | ... | ... |

## Table LXXIX.—Thicknesses of Copper Pipes (L.S.G.).

(May be taken as giving thoroughly reliable strengths for all ordinary marine work.)

| Diameter of pipe in inches. | Steam pipes. | | | | | | Auxiliary exhaust pipes. | Waste steam pipes. | Main water pipes. | Bilge suction and disch. feed suction and fire service. |
|---|---|---|---|---|---|---|---|---|---|---|
| | Boiler pressures in lbs. | | | | | | | | | |
| | 200 | 180 | 155 | 125 | 85 | 50 | | | | |
| 22 | ... | ... | ... | ... | 0 | 7 | ... | ... | 4 | ... |
| 21 | ... | ... | ... | ... | 1 | 7 | ... | ... | 4 | ... |
| 20 | ... | ... | ... | ... | 1 | 7 | ... | ... | 5 | ... |
| 19 | ... | ... | ... | ... | 2 | 8 | .. | ... | 5 | ... |
| 18 | ... | ... | ... | 4/0 | 2 | 8 | .. | 13 | 5 | ... |
| 17 | ... | ... | 6/0 | 3/0 | 3 | 8 | ... | 13 | 6 | ... |
| 16 | ... | 7/0 | 5/0 | 2/0 | 3 | 8 | ... | 14 | 6 | ... |
| 15 | ... | 6/0 | 4/0 | 0 | 4 | 8 | ... | 14 | 7 | ... |
| 14 | 7/0 | 5/0 | 3/0 | 0 | 4 | 9 | 11 | 14 | 7 | ... |
| 13 | 6/0 | 4/0 | 2/0 | 1 | 5 | 9 | 11 | 14 | 7 | ... |
| 12 | 5/0 | 3/0 | 0 | 2 | 5 | 9 | 11 | 14 | 8 | ... |
| 11 | 4/0 | 2/0 | 1 | 3 | 6 | 9 | 11 | 15 | 8 | ... |
| 10 | 3/0 | 0 | 1 | 3 | 6 | 10 | 11 | 15 | 9 | ... |
| 9 | 2/0 | 1 | 2 | 4 | 7 | 10 | 11 | 15 | 9 | ... |
| 8 | 1 | 2 | 3 | 5 | 8 | 10 | 12 | 15 | 9 | ... |
| 7 | 2 | 3 | 4 | 6 | 8 | 11 | 12 | 15 | 10 | 9 |
| 6 | 3 | 4 | 5 | 7 | 9 | 11 | 12 | 15 | 10 | 10 |
| 5 | 5 | 6 | 6 | 8 | 9 | 11 | 12 | 15 | 11 | 10 |
| 4 | 6 | 7 | 7 | 9 | 10 | 12 | 12 | 15 | 11 | 11 |
| 3 | 8 | 9 | 9 | 10 | 11 | 12 | 12 | 15 | 12 | 11 |
| 2 | 10 | 11 | 11 | 11 | 12 | 12 | 12 | 15 | 12 | 12 |
| 1 | 13 | 13 | 13 | 13 | 13 | 13 | 13 | 15 | 13 | 13 |

| Diameter of pipe in inches. | Main eduction and air-pump suction. |
|---|---|
| 35 | 3 |
| 33 | 3 |
| 32 | 4 |
| 29 | 4 |
| 28 | 5 |
| 25 | 5 |
| 24 | 6 |
| 21 | 6 |
| 20 | 7 |
| 17 | 7 |
| 16 | 8 |
| 15 | 8 |
| 14 | 9 |
| 13 | 9 |
| 12 | 10 |
| 11 | 10 |
| 10 | 11 |
| 9 | 11 |
| 8 | 12 |
| 7 | 12 |
| 6 | 13 |
| 5 | 13 |

| Blow-off and scum pipes. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Diameter of pipe in inches, . . | 1 | 1¼ | 1½ | 1¾ | 2 | 2¼ | 2½ | 2¾ | 3 |
| Thickness, L.S.G. | 10 | 10 | 10 | 9 | 9 | 8 | 8 | 7 | 7 |

Feed discharge pipes to be as steam pipes for 30 per cent. higher pressure; but in no case to be taken lower than 125 lbs.

Receiver pipes to be as steam pipes for half the test pressure of the cylinder to which they lead steam; but in no case to be taken lower than 50 lbs.

The above gauges refer to straight pipes only; bends to be suitably strengthened.

## Table LXXX.—Solid-drawn Copper Pipes: Working Pressures allowed in them by the Rule of B.M.E.D. & C. Committee.

| Diameter of Bore. | Thickness in Legal Standard Wire Gauge. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 18. | 17. | 16. | 15. | 14. | 13. | 12. | 11. | 10. | 9. | 8. | 7. | 6. | 5. |
| ins. | | | | | | | | | | | | | | |
| 1·0 | 108 | 156 | 204 | 252 | 300 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 1·25 | 86 | 124 | 163 | 201 | 240 | 297 | ... | ... | ... | ... | ... | ... | ... | ... |
| 1·5 | 72 | 104 | 136 | 168 | 200 | 248 | 296 | ... | ... | ... | ... | ... | ... | ... |
| 1·75 | 62 | 89 | 116 | 144 | 172 | 212 | 253 | 295 | ... | ... | ... | ... | ... | ... |
| 2·0 | ... | 78 | 102 | 126 | 150 | 186 | 222 | 258 | 294 | ... | ... | ... | ... | ... |
| 2·25 | ... | 69 | 91 | 112 | 133 | 165 | 198 | 228 | 260 | 304 | ... | ... | ... | ... |
| 2·5 | ... | 62 | 81 | 101 | 120 | 152 | 177 | 206 | 235 | 273 | 312 | ... | ... | ... |
| 2·75 | ... | ... | 74 | 91 | 109 | 135 | 161 | 187 | 213 | 248 | 283 | 319 | ... | ... |
| 3·0 | ... | ... | 68 | 84 | 100 | 124 | 148 | 172 | 196 | 228 | 260 | 292 | 324 | ... |
| 3·25 | ... | ... | 62 | 77 | 92 | 114 | 136 | 158 | 180 | 209 | 239 | 269 | 299 | ... |
| 3·5 | ... | ... | ... | 72 | 85 | 106 | 127 | 147 | 168 | 195 | 223 | 250 | 277 | 312 |
| 3·75 | ... | ... | ... | 67 | 80 | 99 | 119 | 138 | 156 | 180 | 208 | 233 | 259 | 291 |
| 4·0 | ... | ... | ... | 63 | 75 | 93 | 111 | 129 | 147 | 171 | 195 | 219 | 243 | 273 |
| 4·25 | ... | ... | ... | ... | 70 | 85 | 104 | 121 | 138 | 164 | 184 | 206 | 228 | 257 |
| 4·5 | ... | ... | ... | ... | 67 | 80 | 98 | 114 | 130 | 152 | 173 | 194 | 216 | 242 |
| 5·0 | ... | ... | ... | ... | 60 | 75 | 89 | 103 | 117 | 137 | 156 | 175 | 194 | 218 |
| 5·5 | ... | ... | ... | ... | ... | 68 | 81 | 94 | 107 | 124 | 142 | 159 | 176 | 198 |
| 6·0 | ... | ... | ... | ... | ... | 62 | 74 | 86 | 96 | 116 | 130 | 146 | 162 | 182 |
| 6·5 | ... | ... | ... | ... | ... | ... | 68 | 80 | 89 | 107 | 120 | 134 | 149 | 168 |
| 7·0 | ... | ... | ... | ... | ... | ... | 63 | 74 | 82 | 100 | 112 | 125 | 139 | 156 |
| 7·5 | ... | ... | ... | ... | ... | ... | ... | 69 | 77 | 93 | 104 | 117 | 129 | 145 |
| 8·0 | ... | ... | ... | ... | ... | ... | ... | 64 | 72 | 87 | 98 | 109 | 121 | 136 |

*N.B.*—For the working pressure of brazed pipes multiply by 0·8 or divide by 1·25.

F the diameter of flange for a pipe whose bore is D and has bolts $d$ in diameter.

$F = D + 5d$.

Diameter of bolt circle $D + 4d$.

Thickness of flange in 32nds of an inch $= 0·55\sqrt{F \times \text{W. Press.}} + 6$.

The thicknesses of wrought-iron lap-welded pipes requisite to comply with the B.M.E.D. & C. Committee rule are shown in the following Table:—

### Table LXXXI.—Welded Iron and Steel Pipes (v. page 281): Working Pressure allowed in them by the Rule of the B.M.E.D. & C. Committee.

| Diameter of Bore. | Thickness in 100ths of an inch. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 18. | 20. | 22. | 24. | 26. | 28. | 30. | 32. | 34. | 36. | 38. | 40. | 42. | 44. |
| ins. | | | | | | | | | | | | | | |
| 5·0 | 144 | 180 | 216 | 252 | 288 | 324 | ... | ... | ... | ... | ... | ... | ... | ... |
| 5·5 | 130 | 173 | 206 | 239 | 271 | 304 | 337 | ... | ... | ... | ... | ... | ... | ... |
| 6·0 | 120 | 150 | 180 | 210 | 240 | 270 | 300 | 330 | ... | ... | ... | ... | ... | ... |
| 6·5 | 110 | 138 | 165 | 193 | 220 | 248 | 276 | 303 | 331 | ... | ... | ... | ... | ... |
| 7·0 | 103 | 128 | 154 | 180 | 205 | 231 | 257 | 283 | 308 | 334 | ... | ... | ... | ... |
| 7·5 | 96 | 120 | 144 | 168 | 192 | 216 | 240 | 264 | 288 | 312 | 336 | ... | ... | ... |
| 8·0 | 90 | 112 | 135 | 157 | 180 | 202 | 225 | 247 | 270 | 292 | 315 | 337 | ... | ... |
| 8·5 | 84 | 106 | 127 | 148 | 169 | 190 | 211 | 233 | 254 | 275 | 296 | 317 | 339 | ... |
| 9·0 | 80 | 100 | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 260 | 280 | 300 | 320 | 340 |
| 9·5 | 75 | 94 | 113 | 132 | 151 | 170 | 189 | 208 | 227 | 246 | 265 | 284 | 300 | 319 |
| 10 | ... | 90 | 108 | 126 | 144 | 162 | 180 | 198 | 216 | 234 | 252 | 270 | 288 | 306 |
| 11 | ... | 82 | 98 | 114 | 130 | 147 | 163 | 180 | 196 | 212 | 229 | 245 | 262 | 278 |
| 12 | ... | 75 | 90 | 105 | 120 | 135 | 150 | 165 | 180 | 195 | 210 | 225 | 240 | 255 |
| 13 | ... | ... | 83 | 96 | 110 | 124 | 138 | 152 | 166 | 180 | 194 | 208 | 220 | 235 |
| 14 | ... | ... | 77 | 90 | 102 | 115 | 128 | 141 | 154 | 167 | 180 | 193 | 205 | 218 |
| 15 | ... | ... | 72 | 84 | 96 | 108 | 120 | 132 | 144 | 156 | 168 | 180 | 192 | 204 |
| 16 | ... | ... | ... | 78 | 90 | 101 | 112 | 123 | 134 | 146 | 157 | 169 | 180 | 190 |
| 17 | ... | ... | ... | 74 | 85 | 95 | 106 | 116 | 127 | 138 | 148 | 159 | 169 | 180 |
| 18 | ... | ... | ... | ... | 80 | 90 | 100 | 110 | 120 | 130 | 140 | 150 | 160 | 170 |
| 19 | ... | ... | ... | ... | 76 | 85 | 94 | 104 | 113 | 123 | 133 | 142 | 150 | 160 |
| 20 | ... | ... | ... | ... | ... | 81 | 90 | 99 | 108 | 117 | 126 | 135 | 143 | 153 |
| 21 | ... | ... | ... | ... | ... | 77 | 85 | 94 | 103 | 111 | 120 | 129 | 136 | 145 |
| 22 | ... | ... | ... | ... | ... | ... | 82 | 90 | 98 | 106 | 115 | 123 | 131 | 139 |
| 23 | ... | ... | ... | ... | ... | ... | 78 | 86 | 94 | 101 | 110 | 118 | 126 | 133 |
| 24 | ... | ... | ... | ... | ... | ... | 75 | 83 | 91 | 97 | 105 | 113 | 120 | 128 |

$F$ the diameter of flanges for these pipes whose bore is $D$ and the diameter of the jointing bolts is $d$.

$F = D + 6d$.

Diameter of bolt circle $= D + 4d$.

Thickness of flange in 32nds of an inch $= 0{\cdot}55\sqrt{F \times \text{W. Press.}} + 8$.

## Table LXXXII.—Flanges for Copper Pipes.

| Diameter of pipe in inches. | Diameter of flange. | Thickness of flange. | | Diameter of bolts. | Radius of bolt circle. | Number of bolts. | | | Pitch of bolts. (nearest sixteenth.) | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Steam, feed, blow-off, &c. | Auxiliary exhaust, water pipes, waste and eduction, &c. | | | Steam, feed, blow-off, &c. | Auxiliary exhaust and water pipes. | Waste and eduction pipes. | Steam, feed, blow-off, &c. | Auxiliary exhaust and water pipes. | Waste and eduction pipes. |
| ins. | ins. | in. | in. | in. | ins. | | | | ins. | ins. | ins. |
| 1 | 4 | ½ | ⅜ | ½ | 1⅜ | 4 | 4 | 3 | $1\frac{15}{16}$ | $1\frac{15}{16}$ | 2⅜ |
| 1¼ | 4¼ | " | " | " | 1½ | 4 | 4 | 3 | 2⅛ | 2⅛ | 2⅝ |
| 1½ | 4½ | $\frac{9}{16}$ | $\frac{7}{16}$ | " | 1⅝ | 4 | 4 | 4 | $2\frac{5}{16}$ | $2\frac{5}{16}$ | $2\frac{5}{16}$ |
| 1¾ | 4¾ | " | " | " | 1¾ | 5 | 4 | 4 | $2\frac{1}{16}$ | 2½ | 2½ |
| 2 | 5 | " | " | " | 1⅞ | 5 | 5 | 4 | $2\frac{3}{16}$ | $2\frac{3}{16}$ | 2⅝ |
| 2¼ | 5⅞ | ⅝ | ½ | ⅝ | 2¼ | 5 | 5 | 4 | 2⅝ | 2⅝ | $3\frac{3}{16}$ |
| 2½ | 6⅛ | " | ". | " | 2⅜ | 6 | 5 | 4 | 2⅜ | $2\frac{13}{16}$ | 3⅜ |
| 2¾ | 6⅜ | " | " | " | 2½ | 6 | 5 | 4 | 2½ | $2\frac{15}{16}$ | 3½ |
| 3 | 6⅝ | $\frac{11}{16}$ | $\frac{9}{16}$ | " | 2⅝ | 6 | 5 | 5 | 2⅝ | $3\frac{1}{16}$ | $3\frac{1}{16}$ |
| 3¼ | 6⅞ | " | " | " | 2¾ | 7 | 6 | 5 | 2⅜ | 2¾ | 3¼ |
| 3½ | 7⅛ | " | " | " | 2⅞ | 7 | 6 | 5 | 2½ | 2⅞ | 3⅜ |
| 3¾ | 7⅜ | " | " | " | 3 | 7 | 6 | 5 | 2⅝ | 3 | 3½ |
| 4 | 7⅝ | ¾ | ⅝ | " | 3⅛ | 8 | 6 | 6 | 2⅜ | 3⅛ | 3⅛ |
| 4½ | 8⅛ | " | " | " | 3⅜ | 8 | 7 | 6 | $2\frac{9}{16}$ | $2\frac{15}{16}$ | 3⅜ |
| 5 | 8⅝ | " | " | " | 3⅝ | 9 | 7 | 7 | 2½ | 3⅛ | 3⅛ |
| 5½ | 9⅛ | " | " | " | 3⅞ | 9 | 8 | 7 | 2⅝ | $2\frac{15}{16}$ | 3⅜ |
| 6 | 10⅜ | $\frac{13}{16}$ | $\frac{11}{16}$ | ¾ | 4⅜ | 9 | 7 | 6 | 3 | $3\frac{13}{16}$ | 4⅜ |
| 6½ | 10⅞ | " | " | " | 4⅝ | 9 | 7 | 6 | $3\frac{3}{16}$ | 4 | 4⅝ |
| 7 | 11⅜ | " | " | " | 4⅞ | 10 | 7 | 6 | 3 | $4\frac{3}{16}$ | 4⅞ |
| 7½ | 11⅞ | " | " | " | 5⅛ | 10 | 8 | 7 | $3\frac{3}{16}$ | $3\frac{15}{16}$ | $4\frac{7}{16}$ |
| 8 | 12⅜ | ⅞ | ¾ | " | 5⅜ | 11 | 8 | 7 | $3\frac{1}{16}$ | 4⅛ | 4⅝ |
| 8½ | 12⅞ | " | " | " | 5⅝ | 11 | 9 | 7 | $3\frac{3}{16}$ | 3⅞ | 4⅞ |
| 9 | 13⅜ | " | " | " | 5⅞ | 12 | 9 | 8 | $3\frac{1}{16}$ | 4 | 4½ |
| 9½ | 13⅞ | " | " | " | 6⅛ | 12 | 9 | 8 | $3\frac{3}{16}$ | $4\frac{3}{16}$ | $4\frac{11}{16}$ |
| 10 | 14⅜ | " | " | " | 6⅜ | 13 | 10 | 9 | $3\frac{1}{16}$ | $3\frac{15}{16}$ | 4⅜ |
| 11 | 16¼ | $\frac{15}{16}$ | $\frac{13}{16}$ | ⅞ | $7\frac{3}{16}$ | 13 | 10 | 9 | $3\frac{7}{16}$ | 4½ | 5 |
| 12 | 17¼ | " | " | " | $7\frac{11}{16}$ | 14 | 11 | 10 | $3\frac{7}{16}$ | 4⅜ | 4⅞ |
| 13 | 18¼ | " | " | " | $8\frac{3}{16}$ | 16 | 12 | 10 | $3\frac{5}{16}$ | $4\frac{5}{16}$ | 5⅛ |
| 14 | 19¼ | " | " | " | $8\frac{11}{16}$ | 16 | 13 | 11 | $3\frac{7}{16}$ | $4\frac{3}{16}$ | 5 |
| 15 | 20¼ | " | " | " | $9\frac{3}{16}$ | 17 | 14 | 12 | 3⅜ | 4⅛ | 4⅞ |
| 16 | 21½ | 1 | ⅞ | " | $9\frac{13}{16}$ | 18 | 14 | 12 | $3\frac{7}{16}$ | 4⅜ | 5⅛ |
| 17 | 22½ | " | " | " | $10\frac{5}{16}$ | 19 | 14 | 12 | $3\frac{7}{16}$ | 4⅝ | 5⅜ |
| 18 | 23½ | " | " | " | $10\frac{13}{16}$ | 20 | 15 | 13 | 3⅜ | 4½ | $5\frac{3}{16}$ |

**Note.**—For pressures between 10 and 300 lbs., pitch of bolts is given by—Pitch = $\frac{10\sqrt[3]{dt^2}}{\sqrt[4]{P}}$ + ·5, where $t$ = thickness of flange, $d$ = dia. of bolt, and P = pressure.

The flanges, of which particulars are given in the above Table, are designed either for coupling copper pipe to copper pipe, or copper pipe to a bronze casting of corresponding strength, as indicated in Fig. 47; for connections with cast-iron pipes or valves the flange must be larger, unless studs are used.

Table LXXXIII. (page 277) gives suitable thicknesses for bronze pipes, elbows, T pieces, &c., and will be of assistance in designing bronze stop-valves, expansion stuffing boxes, &c.

The basis and method of construction of the Table are fully explained in the following memoranda:—

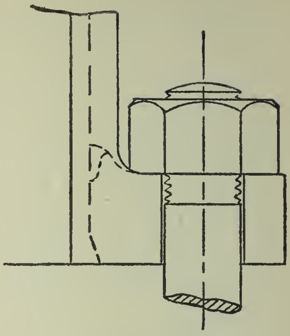

FIG. 47.

**Memoranda.**—H.P. includes feed and bilge delivery, blow-off, scum, steam, fire service, and all sea valves.

L.P. includes all suctions, water pipes not under pressure, exhaust, and waste steam.

The necks of all boiler fittings should be $\frac{1}{16}$ inch thicker than H.P. list.

For plain pipes or cylinders the list is calculated for working pressures of 180 lbs. and 90 lbs. per square inch, for the H.P. and L.P. columns respectively; the stress per square inch of material is taken at 2240 lbs., and a constant addition of ·05 inch is made to cover errors in casting; the formulæ are therefore $t = \cdot 08$ rads. + ·05 inch for H.P., and $t = \cdot 04$ rads. + ·05 inch for L.P.

For H.P. T pieces the square of the thickness (before addition of ·05 inch) has been increased by 33 per cent.,—as a T pipe with all three branches of equal diameter is considered to be that amount weaker than the corresponding plain cylinder without branch in side.

No increase is made in the thickness of L.P. T pieces (over plain pipes), as the pressure assumed, viz. 90 lbs., is already high.

The B.M.E.D. & C. Committee deprecate the use of brazed copper pipes and recommend solid drawn of copper 99·25 and arsenic 0·25 to 0·45 per cent.; the hydraulic test pressure should stress the material to 7500 lbs.; no pipe for a working pressure over 180 lbs. should be of copper if above 6-inch bore. Hydraulic tests for steam pipes to be twice and for feed pipes 2·5 times the working pressure. If, when bent, the angle between the parts is less than 150°, the thickness must be 15 per cent. greater than by rule. The radius of curvature at centre line to be not less than two diameters.

When cast-metal pipes are used for branches, bends, &c., the following rule to be observed:—

$$\text{Thickness in 32nds inch} = \frac{\text{working pressure} \times D}{k} + x,$$ D is the bore in inches.

## Table LXXXIII.—Thickness of Bronze and Cast-Steel (over ⅜) Pipes, T Pieces, &c.

| Diameter in inches. | Plain pipes. | | | | T pieces. | | | |
|---|---|---|---|---|---|---|---|---|
| | Calculated thickness. | | Thickness to be. | | Calculated thickness. | | Thickness to be. | |
| | H.P. | L.P. | H.P. | L.P. | H.P. | L.P. | H.P. | L.P. |
| inches. | inch. | inch. | inch. | inch. | inch. | inch. | inch. | inch. |
| 1 | ·09 | ·07 | ¼ | ¼ | ·10 | ·07 | ¼ | ¼ |
| 1¼ | ·10 | ... | ¼ | ¼ | ·11 | ... | ¼ | ¼ |
| 1½ | ·11 | ·08 | ¼ | ¼ | ·12 | ·08 | ¼ | ¼ |
| 1¾ | ·12 | ... | ¼ | ¼ | ·13 | ... | ¼ | ¼ |
| 2 | ·13 | ·09 | ¼ | ¼ | ·14 | ·09 | ¼ | ¼ |
| 2¼ | ·14 | ... | ¼ | ¼ | ·16 | ... | ¼ | ¼ |
| 2½ | ·15 | ·10 | ¼ | ¼ | ·17 | ·10 | ¼ | ¼ |
| 2¾ | ·16 | ... | ¼ | ¼ | ·18 | ... | ¼ | ¼ |
| 3 | ·17 | ·11 | ¼ | ¼ | ·19 | ·11 | ¼ | ¼ |
| 3¼ | ·18 | ... | ¼ | ¼ | ·20 | ... | ¼ | ¼ |
| 3½ | ·19 | ·12 | 5/16 | ¼ | ·21 | ·12 | 5/16 | ¼ |
| 3¾ | ·20 | ... | 5/16 | ¼ | ·22 | ... | 5/16 | ¼ |
| 4 | ·21 | ·13 | 5/16 | ¼ | ·23 | ·13 | 5/16 | ¼ |
| 4½ | ·23 | ·14 | 5/16 | ¼ | ·26 | ·14 | 5/16 | ¼ |
| 5 | ·25 | ·15 | 5/16 | ¼ | ·28 | ·15 | 5/16 | ¼ |
| 5½ | ·27 | ·16 | ⅜ | 5/16 | ·30 | ·16 | ⅜ | 5/16 |
| 6 | ·29 | ·17 | ⅜ | 5/16 | ·33 | ·17 | ⅜ | 5/16 |
| 6½ | ·31 | ·18 | ⅜ | 5/16 | ·35 | ·18 | ⅜ | 5/16 |
| 7 | ·33 | ·19 | ⅜ | 5/16 | ·37 | ·19 | ⅜ | 5/16 |
| 7½ | ·35 | ·20 | ⅜ | 5/16 | ·40 | ·20 | 7/16 | 5/16 |
| 8 | ·37 | ·21 | ⅜ | 5/16 | ·42 | ·21 | 7/16 | 5/16 |
| 8½ | ·39 | ·22 | 7/16 | 5/16 | ·44 | ·22 | 7/16 | 5/16 |
| 9 | ·41 | ·23 | 7/16 | 5/16 | ·46 | ·23 | ½ | 5/16 |
| 9½ | ·43 | ·24 | 7/16 | 5/16 | ·49 | ·24 | ½ | 5/16 |
| 10 | ·45 | ·25 | ½ | 5/16 | ·51 | ·25 | 9/16 | 5/16 |
| 10½ | ·47 | ·26 | ½ | ⅜ | ·53 | ·26 | 9/16 | ⅜ |
| 11 | ·49 | ·27 | ½ | ⅜ | ·56 | ·27 | 9/16 | ⅜ |
| 11½ | ·51 | ·28 | 9/16 | ⅜ | ·58 | ·28 | ⅝ | ⅜ |
| 12 | ·53 | ·29 | 9/16 | ⅜ | ·60 | ·29 | ⅝ | ⅜ |
| 12½ | ·55 | ·30 | 9/16 | ⅜ | ·63 | ·30 | ⅝ | ⅜ |
| 13 | ·57 | ·31 | ⅝ | ⅜ | ·65 | ·31 | 11/16 | ⅜ |
| 13½ | ·59 | ·32 | ⅝ | ⅜ | ·67 | ·32 | 11/16 | ⅜ |
| 14 | ·61 | ·33 | ⅝ | ⅜ | ·70 | ·33 | ¾ | ⅜ |
| 14½ | ·63 | ·34 | 11/16 | ⅜ | ·72 | ·34 | ¾ | ⅜ |
| 15 | ·65 | ·35 | 11/16 | ⅜ | ·74 | ·35 | ¾ | ⅜ |
| 15½ | ·67 | ·36 | 11/16 | ⅜ | ·76 | ·36 | 13/16 | ⅜ |
| 16 | ·69 | ·37 | ¾ | ⅜ | ·79 | ·37 | 13/16 | ⅜ |
| 16½ | ·71 | ·38 | ¾ | 7/16 | ·81 | ·38 | 13/16 | 7/16 |
| 17 | ·73 | ·39 | ¾ | 7/16 | ·83 | ·39 | ⅞ | 7/16 |
| 17½ | ·75 | ·40 | ¾ | 7/16 | ·86 | ·40 | ⅞ | 7/16 |
| 18 | ·77 | ·41 | 13/16 | 7/16 | ·88 | ·41 | ⅞ | 7/16 |
| 18½ | ·79 | ·42 | 13/16 | 7/16 | ·90 | ·42 | 15/16 | 7/16 |

For cast iron (tensile strength 9 tons), $k=200$, $x=6$.
,, cast steel (tensile 28/35 tons), $k=400$, $x=8$.
,, good bronze (tensile 15 tons), $k=225$, $x=4$.

The minimum thickness for cast metals in 32nds $=2{\cdot}5\sqrt{D}+z$. For cast iron $z$ is 4, for steel 6, and for bronze 2.

**Solid-drawn steel pipes** can now be obtained up to 24 inches bore, and welded steel or iron pipes of any diameter in 100ths of an inch. The minimum thickness as prescribed by the B.M.E.D. & C. Committee for all wrought-iron or steel pipes is $5\sqrt{D}+2$. By this rule the limit for a 4-inch pipe is 0·12 inch, and for a 24-inch one 0·45 inch. Such pipes can be made considerably thinner, but with additional cost and difficulty in attaching the flanges, unless the ends are thickened by upsetting, as with stay tubes in boilers.

**Welded steel or iron tubes** are now also used for steam pipes, and provided they are made of a material which can be depended on to readily and certainly weld, there is no need for cover straps over the weld. For such a purpose steel with a tensile not exceeding 24 tons, with an extension not less than 30 per cent., and with a low sulphur content, should be used.

**The flanges** may be screwed on to the pipe or shrunk on and riveted; they may also be expanded into the flanges, having recesses turned on their inside to receive the impressed metal. Flanges are now often electrically welded to the pipes, and the welding is now so well done as to command confidence when proper materials are used.

**The Board of Trade rules** were as follows :—

170. The working pressure of well-made copper pipes, when the joints are brazed, is found by the following formula :—

$$\frac{6000\times(T-\frac{1}{16})}{D}=\text{working pressure};$$

where $T=$ thickness, in inches ; and
$D=$ inside diameter, in inches.

When the pipes are solid-drawn and not over 10 inches in diameter, substitute in the foregoing formula $\frac{1}{32}$ for $\frac{1}{16}$.

171. The internal pressure on lap-welded wrought-iron pipes made of good material, and on lap-welded or solid-drawn steel pipes, may be determined by the following formula, provided that the minimum thickness is not less than $\frac{1}{4}$ inch, and the workmanship, hydraulic test, &c., are satisfactory :—

$$\frac{6000\times T}{D}=\text{working pressure};$$

where $T=$ thickness, in inches ; and
$D=$ inside diameter, in inches.

Steel pipes which have welded longitudinal seams, and are subject to internal pressure, should in each case be provided with a strap fitted externally and riveted over the weld. All steel pipes should be efficiently annealed after being heated locally for welding or bending them to shape.

The flanges of wrought-iron and of steel pipes should be made of solid wrought material of ductile quality.

Pipes for superheated steam should be made of wrought-iron or steel, and not of copper.

172. Efficient means should be provided for draining all steam pipes. Boiler stop-valves cannot be regarded as suitable for this purpose. All drain cocks or valves should be accessible, and so placed as to render it practicable to drain the water from any portion of the steam pipes or chests in connection therewith. Drain pipes should be fitted to drain cocks or valves when the latter are in such a position that the water or steam discharged from them would be likely to cause personal injury. It is desirable that the drains should be automatic in their action.

173. The parts of a socket expansion joint subject to rubbing action should be made of brass or of other metal which will not rust.

In all cases in which such a joint is fitted to a bent steam pipe, the Surveyor should require a fixed gland and bolts, or other efficient means, to be provided to prevent the end of the pipe being forced out of the socket. This regulation should be complied with in all cases of bent pipes fitted with socket expansion joints, and it is also desirable that fixed glands and bolts should be fitted to the expansion joints of *straight* steam pipes, as cases have occurred, particularly with small straight pipes, in which the ends have been forced out of the sockets.

A socket expansion joint on a bent pipe is not a desirable arrangement, and, when adopted, the pipe should be anchored or provided with a strut at the bend, to relieve it of any undue bending stresses which might otherwise be produced by the internal pressure on a surface of the pipe equal to the area due to its bore.

174. Surveyors should pay particular attention to the examination and testing of steam pipes, and a record of the tests should be kept in the office boiler-book.

Tables LXXX. and LXXXI. are calculated in accordance with the rules given in paragraphs 170 and 171 of the above rules.

**Pipe arrangements.**—The greatest care should be taken in scheming arrangements of steam and exhaust pipes to keep them as far as possible in the same horizontal plane ; a "pocket" in which water can collect must never be permitted under any circumstances. Neither should a steam pipe which has been led horizontally for some distance be suddenly bent up into the vertical ; *all* sharp bends are sources of danger and to be avoided, but sharp rising bends are specially so.

The provision of proper arrangements for draining all steam pipes, regulator valve boxes, shut-off valve boxes, &c., is also a matter requiring the closest attention—as water that has once left the boilers and entered the steam pipes will never, with any ordinary arrangement of pipes, drain back to the boiler in the face of the issuing steam. Where the pipes are long it is very necessary to provide collectors of some sort—as traps do not act with the necessary rapidity; the inlet pipes to the collectors should be large, and so placed as to arrest and lead away the rapidly moving water.

The question of the arrangements necessary to allow for the expansion of such pipes as vary much in temperature is a difficult one, and can only be very generally treated here.

Steam pipes for high pressures are so thick and rigid that ordinary bends are of little or no use for giving elasticity, and stuffing boxes should be provided wherever expansion and contraction will take place in ordinary work.

If expansion stuffing boxes and glands are not entirely of bronze, as is usual in Naval work, the glands should be of that metal, and the bodies so bushed or lined with it as to prevent the possibility of rusting up and consequent jamming.

These stuffing boxes should be cast with the necessary flanges or brackets to secure them to the bulkheads or other fixed bases, and the end of each length of piping (the end furthest away from the stuffing box) should be anchored in a similar way.

In the best class of work the end of the pipe that enters the stuffing box is always made of a separate piece in cast bronze, and this method has the advantage that the pair of flanges at the junction between the bronze pipe and the copper or iron one form a convenient "guard-flange," or flange to take the "guard" bolts, which prevent the pipe from being blown out of the stuffing box. These guard bolts should never be omitted except the pipe be very short and rigid, and the end attachments also very rigid—as fatal accidents have occurred in the past owing to their absence.

With the high pressures and temperatures now obtaining the faucet joint does not provide for the expansion of steam pipes inasmuch as if the packing is steamtight the friction is sufficient to grip the pipe. A more satisfactory arrangement is to have a carefully turned spigot whose end is bored so as to be thin enough to expand somewhat under pressure in the smoothly bored socket without any stuffing box and gland. This permits of the free expansion of the pipes without any leakage of steam or water. Both spigot and socket should be of incorrodable metals preferably of different compositions to avoid seizing.

It is also very important that those in charge of the fitting-up of the pipes on board ship should have full and clear information (a special small scale tracing or diagram is best) as to the amount of expansion anticipated, and to be provided for, at each stuffing box.

## Steel Steam Pipes.

For steam pressures over 180 lbs. per square inch steel pipes are now generally used. Up to 6 inches diameter they should be solid drawn, and above that they may be welded; they are better, however, solid drawn, as generally supplied for Naval work, and though somewhat more costly they are much lighter.

**Solid-drawn steel tubes**, without seam or weld, are now made in all sizes up to 24 inches diameter, and can be obtained (Chesterfield Tube Co.) of a minimum thickness equal to 0·015 of the diameter; they are usually made of special steel having an ultimate tensile strength of about 27 tons, with an elongation of 35 per cent. in 2 inches; but they can be made of a stronger material if required. Such tubes are finished by cold drawing, and when so very thin are, of course, very expensive; but when, in accordance with the following rules, which hold good for steam and all other pipes exposed to pressure, the cost is moderate, especially considering their safety and lightness.

The diameter is D, the length L, both in inches; the pressure is P in pounds per square inch:—

**Rule 235.** **Thickness of steam cold solid-drawn pipes in ins.**

$$= \frac{D \times P}{12{,}000} + 0{\cdot}08,$$

or **thickness of steam cold solid-drawn pipes in 64ths of an in.** $= \frac{D \times P}{185} + 5.$

**Thickness of hot drawn in 64ths of an in.**

$$= \frac{D \times P}{185} + 8.$$

*Pipes over* 6 *inches in diameter* may be made by welding the joint, as with boiler tubes; and in the somewhat larger sizes, connecting them with a butt joint with a single strap single-riveted; without welding they should be double-riveted. The following rule holds good for such sorts:—

**Rule 236.**

**Thickness of welded, &c., steam pipes in 64ths of an in.** $= \frac{D \times P}{140} + 7.$

*Exhaust pipes* subject to external pressure of 15 lbs., continuously or temporarily, may be solid drawn, welded, or riveted of a thickness given by the following:—

**Rule 237.**

**Thickness of exhaust pipes in ins.** $= \frac{\sqrt{D \times L}}{384} + 0{\cdot}063,$

or ,, ,, **in 64ths of an in.** $= \frac{\sqrt{D \times L}}{6} + 4.$

Tables LXXXIV., LXXXV., and LXXXVI. give the thicknesses of the various pipes in 64ths of an inch, as they can be made, worked, and used in practice. The Admiralty have all such pipes galvanised by the cold process; as a preventive of corrosion, both inside and out, it is well worth the cost. No doubt the zinc will, in course of time, wear away from the inside, but it will always protect the outer surface.

## Table LXXXIV.—Thickness of Cold Solid-drawn Steel Steam and Feed Pipes in 64ths of an inch.

| Diameter of Bore. | Working pressures in lbs. per square inch. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 50 | 60 | 70 | 80 | 100 | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 260 | 280 |
| ins. | | | | | | | | | | | | | | |
| 2·0 | 8 | 8 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 11 | 11 |
| 2·5 | 8 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 12 | 12 |
| 3·0 | 8 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 12 | 12 | 12 |
| 3·5 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 12 | 12 | 12 | 13 |
| 4·0 | 9 | 9 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 12 | 12 | 12 | 13 | 14 |
| 4·5 | 9 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 12 | 12 | 13 | 13 | 13 | 14 |
| 5·0 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 12 | 12 | 13 | 13 | 13 | 14 | 15 |
| 6·0 | 10 | 10 | 10 | 10 | 11 | 11 | 12 | 12 | 13 | 13 | 14 | 14 | 15 | 16 |
| 7·0 | 10 | 10 | 10 | 11 | 11 | 12 | 12 | 13 | 13 | 14 | 15 | 15 | 16 | 17 |
| 8·0 | 11 | 11 | 11 | 11 | 12 | 12 | 13 | 13 | 14 | 15 | 16 | 16 | 17 | 18 |
| 9·0 | 11 | 11 | 11 | 11 | 12 | 13 | 13 | 14 | 15 | 16 | 17 | 17 | 18 | 19 |
| 10 | 12 | 12 | 12 | 12 | 13 | 13 | 14 | 15 | 16 | 17 | 17 | 18 | 19 | 20 |
| 11 | 13 | 13 | 13 | 13 | 14 | 14 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 21 |
| 12 | 15 | 15 | 15 | 15 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 21 | 22 |
| 13 | 16 | 16 | 16 | 16 | 16 | 16 | 17 | 17 | 18 | 19 | 20 | 21 | 22 | 23 |
| 14 | 18 | 18 | 18 | 18 | 18 | 18 | 18 | 18 | 19 | 19 | 21 | 22 | 23 | 24 |
| 15 | 19 | 19 | 19 | 19 | 19 | 19 | 19 | 19 | 20 | 20 | 21 | 22 | 24 | 24 |
| 16 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 20 | 21 | 21 | 22 | 23 | 24 | 25 |

*N.B.*—The pressure in feed pipes will be greater than the boiler pressure, and if from pumps worked by the main engines, should be taken at 1·3 to 1·6 times the working pressure. If from independently worked direct pumps, 1·2 to 1·4 times is ample.

### Table LXXXV.—Thickness of Welded or Riveted Steam Pipes in 64ths of an inch.

| Diameter of Bore. | Working pressure in lbs. per square inch. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 50 | 60 | 70 | 80 | 90 | 100 | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 260 | 280 | 300 |
| ins. | | | | | | | | | | | | | | | | |
| 8 | 13 | 13 | 13 | 14 | 14 | 14 | 15 | 15 | 16 | 16 | 17 | 17 | 18 | 18 | 19 | 19 |
| 9 | 13 | 13 | 13 | 14 | 14 | 14 | 15 | 15 | 16 | 17 | 17 | 18 | 18 | 19 | 19 | 20 |
| 10 | 13 | 13 | 14 | 14 | 14 | 15 | 15 | 16 | 17 | 17 | 18 | 18 | 19 | 20 | 20 | 21 |
| 11 | 13 | 13 | 14 | 14 | 15 | 15 | 16 | 16 | 17 | 18 | 18 | 19 | 20 | 21 | 21 | 22 |
| 12 | 13 | 14 | 14 | 15 | 15 | 15 | 16 | 17 | 18 | 18 | 19 | 20 | 21 | 22 | 22 | 23 |
| 13 | 13 | 14 | 14 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 20 | 21 | 22 | 23 | 24 |
| 14 | 13 | 14 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 19 | 20 | 21 | 22 | 23 | 24 | 25 |
| 15 | 14 | 14 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 |
| 16 | 14 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 |
| 17 | 14 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 25 | 26 | 27 | 28 |
| 18 | 14 | 15 | 16 | 16 | 17 | 17 | 18 | 20 | 21 | 22 | 23 | 24 | 25 | 27 | 28 | 29 |
| 19 | 15 | 15 | 16 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 24 | 25 | 26 | 28 | 29 | 30 |
| 20 | 15 | 15 | 16 | 17 | 17 | 18 | 19 | 21 | 22 | 23 | 24 | 25 | 27 | 28 | 30 | 31 |
| 21 | 15 | 16 | 16 | 17 | 18 | 18 | 20 | 21 | 22 | 24 | 25 | 26 | 28 | 29 | 31 | 32 |
| 22 | 15 | 16 | 17 | 17 | 18 | 19 | 20 | 22 | 23 | 24 | 25 | 27 | 29 | 30 | 32 | 32 |
| 23 | 15 | 16 | 17 | 17 | 18 | 19 | 20 | 22 | 23 | 25 | 26 | 28 | 29 | 30 | 32 | 32 |
| 24 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 23 | 24 | 25 | 26 | 28 | 30 | 31 | 32 | 32 |

Table LXXXVI.—Thickness of Exhaust Pipes (Steel), Solid-drawn or Riveted, in 64ths of an inch.

| Diameter of Bore. | Length of pipe in inches between flanges or ribs. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 | 100 | 110 | 120 | 130 | 140 | 150 |
| ins. | | | | | | | | | | | | | | |
| 6 | 7 | 7 | 8 | 8 | 8 | 8 | 8 | 8 | 8 | 9 | 9 | 9 | 9 | 9 |
| 7 | 7 | 8 | 8 | 8 | 8 | 9 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 |
| 8 | 8 | 8 | 8 | 9 | 9 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 | 10 |
| 9 | 8 | 8 | 9 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 10 |
| 10 | 8 | 8 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 11 | 11 |
| 11 | 8 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 11 | 11 | 11 |
| 12 | 9 | 9 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 |
| 13 | 9 | 9 | 9 | 9 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 | 12 |
| 14 | 9 | 9 | 10 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 | 12 | 12 |
| 15 | 9 | 9 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 | 12 | 12 | 12 |
| 16 | 10 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 |
| 17 | 10 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 13 |
| 18 | 10 | 10 | 10 | 10 | 11 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 13 |
| 19 | 10 | 10 | 11 | 11 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 13 | 13 |
| 20 | 10 | 10 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 12 | 13 | 13 | 13 |
| 22 | 10 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 13 | 13 | 13 | 13 | 14 |
| 24 | 11 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 13 | 13 | 13 | 14 | 14 | 14 |
| 26 | 11 | 11 | 11 | 12 | 12 | 12 | 12 | 13 | 13 | 13 | 14 | 14 | 14 | 15 |
| 28 | 11 | 11 | 12 | 12 | 12 | 12 | 13 | 13 | 13 | 14 | 14 | 14 | 15 | 15 |
| 30 | 11 | 12 | 12 | 12 | 12 | 13 | 13 | 13 | 13 | 14 | 14 | 15 | 15 | 15 |
| 32 | 12 | 12 | 12 | 12 | 13 | 13 | 13 | 13 | 14 | 14 | 15 | 15 | 15 | 16 |
| 34 | 12 | 12 | 13 | 13 | 13 | 13 | 14 | 14 | 14 | 15 | 15 | 15 | 16 | 16 |
| 36 | 12 | 13 | 13 | 13 | 13 | 14 | 14 | 14 | 14 | 15 | 15 | 16 | 16 | 17 |

The thicknesses are for straight pipes only; bends to be suitably strengthened.

Welded pipes should be subjected to a water-pressure test of about three times the working pressure, and should be smartly struck with a steel hand hammer all along the weld while the pressure is on. The Admiralty at present require all welded pipes to be fitted with an external single-riveted butt-strap over the weld, but some of our most experienced tube makers are more than doubtful of the value of this addition.

Solid-drawn pipes up to about 5 inches diameter may be readily bent to moderate curves by filling with lead and bending cold in a press, as is usual with copper pipes. Lap-welded iron pipes of 3, 4, and 5 inches diameter, galvanised externally, are largely used in H.M. Navy for bilge suctions and connections, and are bent by filling with sand and heating to a low red heat. The usual thickness of these pipes is $\frac{1}{16}$ inch per inch of diameter.

The flanges for steel pipes should be stamped from steel plates, and may be made with a small bead on the back for such sizes as are screwed on to the pipes, and of angle bar section for those that are to be riveted on. For the smaller sizes, say up to 5-inch pipes, a good plan is to cut a half thread on the inside of the flange and then expand the pipe against the surface so grooved.

## Tests of Steam Pipes (Solid-drawn Steel).

The present Admiralty requirements as regards steel steam pipes are as follows:—

To be made from acid open-hearth steel, which must stand the following tests:—

Annealed strips cut lengthwise from each pipe must have a tensile strength of between 24 and 27 tons, and must show an elongation of 33 per cent. in 2 inches before fracture.

Strips cut either way, 1½ inches wide, heated to a low cherry red and quenched in water at 82° F., must stand bending double in a press, over a radius equal to 1½ times thickness of strip tested, without fracture.

Pieces 2 inches long cut from ends of pipes must stand hammering down endwise, when cold, until length is reduced to 1 inch, and flattening until sides are within twice the thickness of the material of each other, without signs of fracture. Pipes intended to carry 300 lbs. steam must also, after pickling, and after fitting and facing of flanges, but before galvanising externally, stand satisfactorily a water-pressure test of 600 lbs. per square inch.

## Steam Pipes (Welded Steel).

To be made from acid open-hearth steel.

All welding to be by gas on a mandril.

Each welded joint to be covered by an external single-riveted butt-strap at least $\frac{1}{16}$ inch thicker than pipe.

Edges of strap to be bevelled to an angle of 75°.

Tensile and bending tests as for solid-drawn pipes (above).

The plates are, in addition, to stand such other forge tests, hot and cold, as the overseer may consider necessary, and pieces are to be welded and broken in the testing machine; also some pieces cut from the pipes are to be broken to ascertain the efficiency of the welding.

Pipes of this construction must also stand the water-pressure test specified above for solid-drawn pipes.

### Flanges of Steel Steam Pipes.

To be forged from the solid, not welded.

To be made from acid open-hearth steel.

Material to have a tensile strength of between 27 and 30 tons per square inch, and to show 27 per cent. elongation in 2 inches before fracture.

Bending tests as for material of pipes (above).

## STOP-VALVES.

These should always be made with the spindle a separate piece, distinct from the valve, and so attached as to allow a little "play,"—thus enabling the valve to accommodate itself to the seat when the latter is slightly "out of truth."

Steam stop-valves, above about 3½ inches diameter, should have external and accessible spindle nuts, carried in crossheads or bridges; for water valves this is not so necessary, but it is usually best to make only one type of valve for both purposes,—only varying the strengths of the parts.

Screw glands should not be used for valves over 1 inch in diameter, and, when used, should always be fitted with some means of locking.

Ordinary gland studs should be fitted with "check" or "lock" nuts and split pins.

Hand-wheels for stop-valves should never be turned and polished; they are better left rough and served with twine.

For large valves (over 10 inches in diameter, say) a forged cross-handle (four arms), with the ends turned over at right angles, either towards or away from the valve, is better than a wheel; many engineers prefer this type of handle to the hand-wheel for all valves above about 4 inches diameter, and there is much to be said in its favour.

For boiler-room valves a similar forged handle, but with only two arms, one bent outwards and the other straight, is the best.

For very large stop-valve chests the best material is, undoubtedly, cast-steel. Trouble is occasionally caused by unsound castings, but this cannot be regarded as unavoidable.

In Naval practice, all stop-valves for water, oil, air, &c., are entirely of bronze, and steam valves of small and moderate sizes are constructed in the same way, whilst the larger sizes have commonly cast-steel chests with bronze seats, &c.

The number of joints may often be considerably reduced by casting the expansion stuffing boxes along with the stop-valve chests.

The handling of large shut-off valves may be much facilitated by the addition of a small by-pass valve.

The whole of the valves in any vessel should open or close by turning the hand-wheel in the same direction. The Admiralty rule is to close with a right-hand motion. To remove all doubt, it is a good plan to cast on the upper surface of all hand-wheels the words "open" and "shut," thus:—OPEN⟵X⟶SHUT.

## THE BALANCING OF ENGINES.

In an unbalanced engine there are two perfectly distinct sets of forces which give trouble; first, those set up by unbalanced rotating parts; and second, those caused by unbalanced reciprocating parts. The effect of an unbalanced rotating part, such us a crank, is to produce a force acting radially from the shaft axis through the centre of gravity of the part, which causes the shaft to spring and its axis to revolve in a small orbit round the centre of rotation, the shaft thus moving eccentrically and tending to grind out the bearing round its whole circumference. This may result in undue wear and tear and heating of bearings; if the bearing surface is ample, the force will tend to spring the engine seatings and cause the whole engine to move with the eccentrically-moving shaft; if the seatings are rigid, the whole hull of the ship may respond to the movement of the engines and thus be caused to vibrate.

The forces set up by vertically reciprocating parts are mostly vertical, and, if their resultant acts through the centre of gravity of the engine, the tendency is to *tilt* the whole engine and ship up and down, but if acting away from that vertical line of the engines they tend to rock the engines. The result in either case, if the vibrations are severe, or the hull is lightly built, is to cause deterioration of the ship's structure, with more or less leakage, and also discomfort to passengers and crew, and in the case of warships to produce faulty gunnery. Even small unbalanced forces may produce great vibration if their periods synchronise with the vibration periods of the hull or any part of it.

Part of the connecting-rod should be considered as reciprocating with the piston, &c., and the other part as revolving with the crank-pin, &c., and the relative magnitudes of these parts may be determined as follows:—Find the centre of gravity of the rod, and if it be found to lie at, say, 70 per cent. of the rod's length from the top end, the reciprocating weight may be taken as 100 – 70, or 30 per cent. of the total weight of rod; and the remaining 70 per cent. must then be treated as revolving with the crank-pin, &c.

Strictly speaking, the force set up by a rotating weight can only be truly balanced, or neutralised, by the similar and equal force set up by another weight placed at the opposite side of the axis and rotating in the same transverse plane; and the force set up by a reciprocating weight can only be neutralised by the similar and equal force set up by another weight reciprocating on the same line, but in the opposite direction.

A single crank-arm balance weight cannot be arranged to rotate with its centre of gravity in the transverse plane through the mid-length of the crank-pin; it is therefore halved and placed one on each arm, symmetrically. Strictly, such balance weights should not have a greater moment (weight × radius of centre of gravity) than the unbalanced rotating parts, for if they are made heavier, with a view to balance also a portion of the reciprocating weights, a horizontal force is set up equal in magnitude to that portion of the vertical inertia force that is balanced.

Still there are cases, such as those of the numerous auxiliary engines in which the reduction of the inertia forces of the reciprocating parts, and the vibration caused thereby, are of greater importance than the creation of small transverse forces; hence, for the generality of such cases this simple method of "overbalancing" serves admirably. A very small weight attached to the rim of a fan, or placed in the crank-disc of a centrifugal pump engine, though not in the transverse plane through the mid-length of crank-pin, yet produces a marked improvement in the running of the single-crank engine, and should be always fitted unless a more perfect arrangement is necessary. Balance weights forged with the crank-arms are, of course, better, but, except in the case of built shafts, they are very much more expensive.

A piston, with rod, &c., cannot usually be balanced by another weight reciprocating in the opposite sense in the same vertical line, but "bob-weights," driven by special eccentrics or cranks, can be symmetrically arranged as regards the transverse plane through the centre-line of piston-rod, and may be on the same side of the crankshaft as the piston, &c., to be balanced; but the extra cost, weight, and complication are serious objections, and have led to the perfecting of what may be called "collective" balancing—a method which secures to the full the advantage of quenching vibration, though leaving the wear and tear of bearings, to some extent at any rate, to be resisted by ample surfaces.

As already remarked, the force set up by an unbalanced rotating weight is always radial and acts from the shaft axis through the centre of gravity of the rotating weight. Its magnitude (neglecting gravity) is given by :—$F = WrN^2 \times {\cdot}00034$,* where W = weight of body in pounds, $r$ = radius of its centre of gravity in feet, and N = number of revolutions per minute. The vertical component of this force for any crank position has the value :—$F_V = WrN^2 \times {\cdot}00034 \times \cos\theta$, where $\theta$ is the angle between crank and engine centre-line; or, if a graphic method is preferred, for $\cos\theta$ read $\frac{AB}{AC}$ as indicated in Fig. 48.

---

* Obtained by substituting $\frac{2\pi r}{60}N$ for V in the formula $F = \frac{WV^2}{gr}$.

This fraction has the value 1 at the top and bottom centres, 0 at the points D and E, and intermediate values at other positions.

Similarly, the horizontal component is :—$F_H = WrN^2 \times \cdot 00034 \times \sin\theta$, or, graphically, for $\sin\theta$ read $\frac{BC}{AC}$. This fraction has the value 1 at positions D and E, and 0 at top and bottom centres.

The force due to an unbalanced weight reciprocating vertically is purely vertical, as previously stated; and it is a maximum at each end of the stroke and falls to zero at a point near mid stroke. Its magnitude and its effect upon the effective pressure diagram (fig. 2), may be determined by Mr Rigg's method as follows :—Describe a circle (fig. 49) with radius equal to $WrN^2 \times \cdot 00034$, to any convenient scale, W being in this case the weight in pounds of the reciprocating parts, which are here considered as being located at the crank-pin and revolving with it. Divide the diameter AB into ten equal parts and re-divide each end part into two. Draw horizontal lines through these division marks to meet the circle as shown. Then—supposing the connecting-rod to be four cranks in length—from C set off CH equal to one-fourth CB; and from F and G set off, in the opposite direction, FJ and GK, each equal to CH; and through J, H, and K strike a circular arc (in this case from point A). Then the distances LM, &c., will give the forces required at each of the thirteen (including the two end points) points in the stroke, and they may be designated − or + according as they are measured above or below arc JHK.

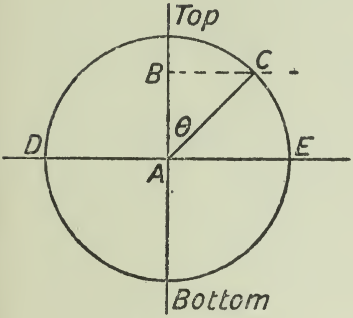

FIG. 48.

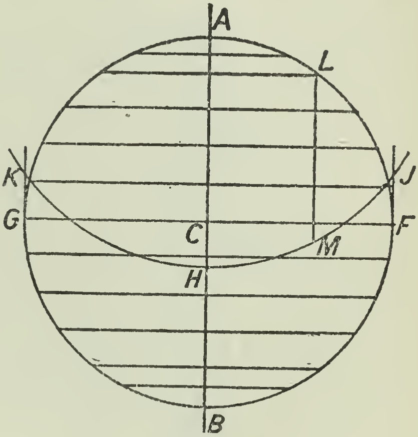

FIG. 49.

Draw a horizontal line AB (fig. 50) to represent the stroke of the piston, and divide it similarly to AB in the previous figure, and at each point set off, up or down, the force given by the ordinate of that figure, divided by the piston area. Then, if a fair curve be drawn through the ends of these ordinates, it will show the variation in the pressure per square inch required to accelerate and retard the piston, &c., during one stroke, if gravity be neglected. The variation during the return stroke is shown by the dotted line. But gravity must be allowed for, and to do this a new base line must be drawn at a distance *w* above AB (or *below* AB for the up stroke), *w* representing to scale the weight of the reciprocating parts per square inch of piston,—and all measurements must then be taken from *ab* in lieu of AB. Now draw the effective pressure diagram (fig. 2), above this figure, and to the same scale, and add to, or subtract from, each of its ordinates the amount indicated by the lower figure; and, through the new points thus found, draw a fair curve as indicated by the dotted line CED. The ordinates from CD to this curve will give the net (except as regards friction losses) effective pressures acting through the piston rods; and from these the net pressures at the crank-pin, or the net twisting moments, may be determined by the method shown in fig. 15, and may be exhibited by a diagram resembling fig. 16, or by the circular form of the same diagram. The ordinates of these figures will then give the net crank-pin pressures, or the net twisting moments, at every point in the crank-pin path, except as regards friction losses and the small fly-wheel effect of the propeller, cranks, &c.

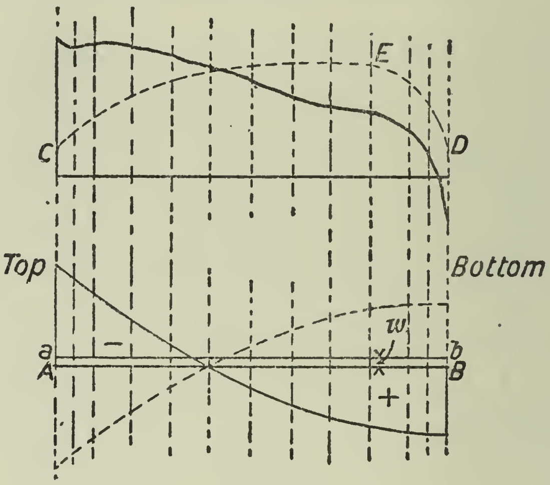

FIG. 50.

The sum of the unbalanced forces set up by any crank and its piston, piston-rod, connecting-rod, &c., at every position during a revolution, may be exhibited by a diagram constructed in the following manner:—

First, find the amount of the radial force set up by the rotating parts, by the method already given, and describe a circle with a radius representing this force to any convenient scale (fig. 51). And secondly, from a selected number of points in the circumference of this circle (which is also to be considered as representing the path of the crank-pin) set off, upwards or downwards, vertical ordinates representing, to the same scale, the forces due to the reciprocating parts (from fig. 65), and through the extremities of these ordinates draw a fair curve. Then for any crank angle, say 30° down from top centre, the total force is indicated in magnitude and direction, by the resultant CB of the two forces CA and AB, and so for other positions of crank.

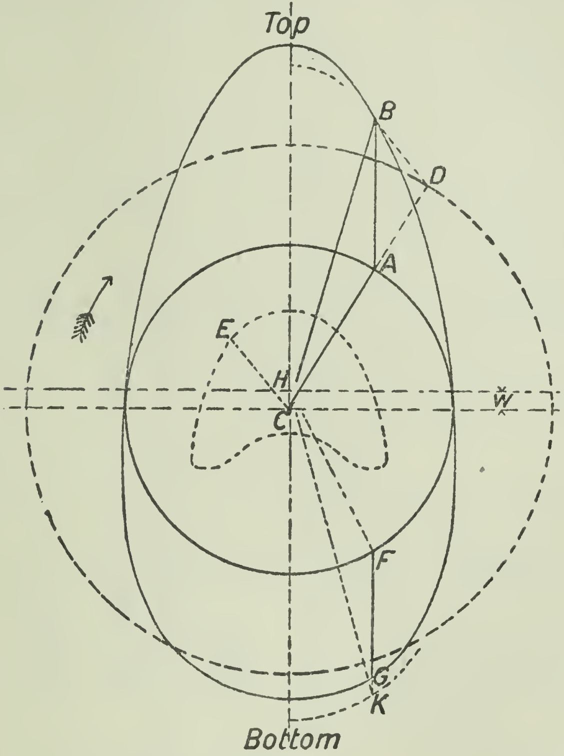

FIG. 51.

If the rotating parts are balanced (either partially, wholly, or "overbalanced"), describe another circle with radius CD representing centrifugal force due to balance-weights (again to same scale) and join DB. Then the resultant DB gives the magnitude and direction of the total force under these conditions for the crank position CD; and if lines such as CE, equal and parallel to the resultants so found, be set off from the centre C, and a fair curve be drawn through their extremities, the new diagram thus produced shows at a glance the variations in the total force during one complete revolution. Note that in this case the lines CE, &c., are set off the reverse way round.

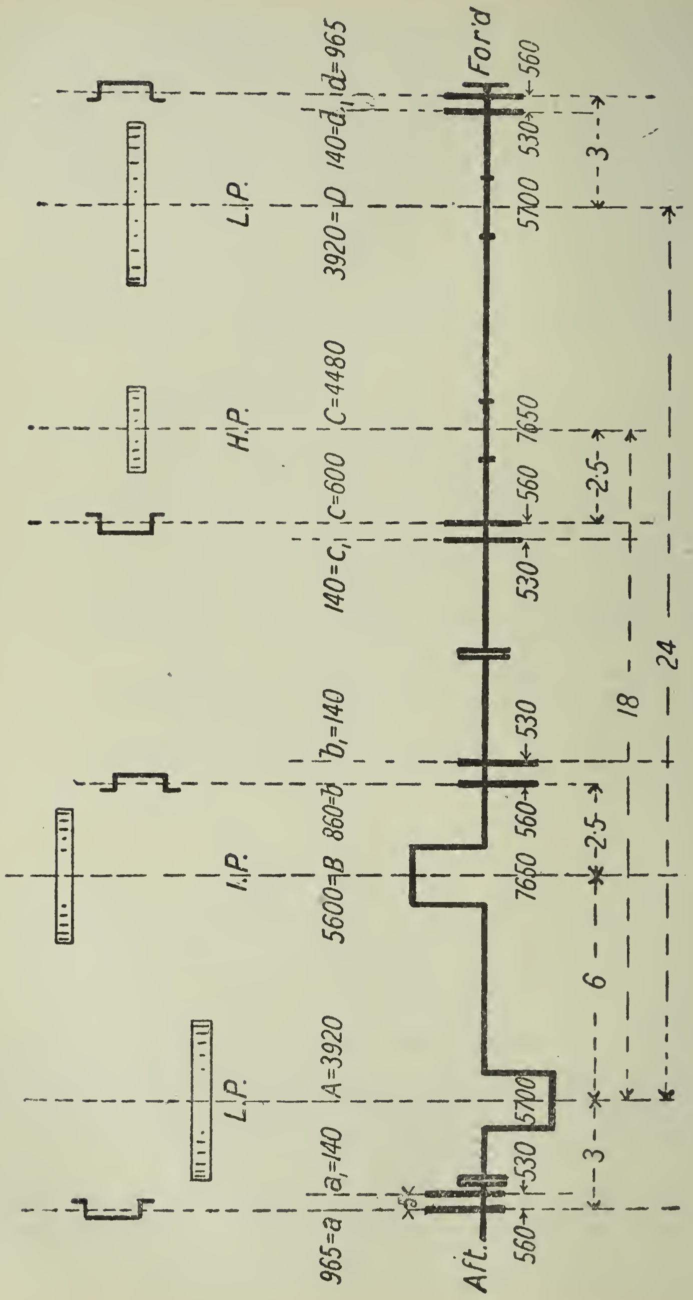
Aft.
965=a
a₁=140
A=3920
L.P.
5600=B
I.P.
860=b
b₁=140
140=c₁
c=600
H.P.
C=4480
3920=D
L.P.
140=d₁
d=965
For'd
560
530
5700
7650
560
530
530
560
7650
5700
530
560
5
3
6
2·5
18
24
2·5
3

Fig. 52.

If gravity is to be allowed for, as it must be when estimating forces acting on main bearings, framings, &c., a new horizontal centre line should be drawn at a distance W above the first one,—where W represents the weight of the rotating parts, to scale; and the ordinates AB, FG, &c., must be respectively shortened and lengthened by an amount representing the weight of the reciprocating parts. All measurements should then be taken from the point H, and HK would be one of the set of resultants corresponding to CB, &c., in the former case, where gravity is not considered.

When vibrations and the forces causing them are alone under consideration, gravity need not be considered, for the reason that the weights of the moving parts may be regarded as producing a slight initial depression of the engine seatings, &c., in addition to that caused by the weights of the stationary parts, and the vertical vibrations may then be considered as taking place equally above and below this lowered position, instead of unequally above and below the higher one.

The object of the "collective" system of balancing (best known as the "Yarrow, Schlick, and Tweedy System") is not to balance the moving parts in connection with each cylinder independently, but so to space the centres of the various cylinders, so to arrange the various crank angles, and so to proportion the weights of the moving parts as to cause the whole of the disturbing forces to cancel out. In order that an engine may be balanced on this system, it must have not less than four cranks, and, generally speaking, engines with fewer cranks must be balanced by some combination of the rotating balance weight and the reciprocating "bob-weight."

As soon as the general design and arrangement and leading dimensions of an engine (say four-crank triple) have been decided upon, its condition as regards balance may be examined by the following approximate method, approximate because it takes no account of disturbances due to obliquity of connecting rod:—

Make a skeleton sketch of the engine (as indicated by fig. 52) and figure upon it the weights of the unbalanced rotating parts referred to the crank-pin $\left(i.e.\ \frac{\text{Weight of part} \times \text{dist. of its c. of g. from shaft axis}}{\text{Radius of crank path}}\right)$, the weights of the reciprocating parts, and the distance of each set of parts from an imaginary transverse plane of reference through the mid-length of the after crank-pin. Then, to examine the forces and moments due to the reciprocating parts, which act in the vertical plane through the shaft axis and tend to set up vertical vibrations, first tabulate the data thus:—

## Table LXXXVIa.

| Marks on Cranks, etc. | Weights of Reciprocating Parts. lbs. | Distances from Ref. Plane to Centres of Piston-rods, etc. ft. | Moments of Reciprocating Parts. ft. lbs. | Weights of Unbalanced Rotating Parts. lbs. | Moments of Rotating Parts. ft. lbs. |
|---|---|---|---|---|---|
| $a$ | 965 | −3 | −2,895 | 560 | −1,680 |
| $a_1$ | 140 | −2·5 | − 350 | 530 | −1,400 |
| A | 3,920 | ... | ... | 5,700 | ... |
| B | 5,600 | 6 | 33,600 | 7,650 | 45,900 |
| $b$ | 860 | 8·5 | 7,310 | 560 | 4,760 |
| $b_1$ | 140 | 9 | 1,260 | 530 | 4,770 |
| $c_1$ | 140 | 15 | 2,100 | 530 | 7,950 |
| $c$ | 600 | 15·5 | 9,300 | 560 | 8,680 |
| C | 4,480 | 18 | 80,640 | 7,650 | 137,700 |
| D | 3,920 | 24 | 94,140 | 5,700 | 136,800 |
| $d_1$ | 140 | 26·5 | 3,710 | 530 | 14,045 |
| $d$ | 965 | 27 | 26,055 | 560 | 15,120 |

In this particular case the stroke of pistons was five times the travel of valves; the weights of valve and gear given in columns two and five are therefore one-fifth of the actual weight. The minus sign is used because the after low-pressure gear is to the left of the reference plane, whilst all other weights are to the right of it. The weights in connection with the after low-pressure crank have no moment about the plane of reference, because the plane passes through their centre of gravity.

As regards valve gear, the valve, rod, block, half the links and drag rods, and a proportion of the eccentric rod and strap (say, four-tenths) are taken as reciprocating over the ahead eccentric; and the remaining half of the links, and a similar proportion of the astern eccentric rod and strap, over the astern eccentric. The unbalanced portions of the sheaves and the remainders of the rods and straps (six-tenths) are taken as rotating parts.

Additions are made to the reciprocating weights to allow for the unbalanced parts of pump levers, links, rods, and buckets.

The obliquity of eccentric rods is usually so small that it may be neglected.

Next assume (as first working hypothesis) that the cranks are arranged with a forward pair, opposite to one another, and an after pair, also opposite to one another, but that the forward pair are at right angles to the after pair. This combination has the advantage of getting the lighter L.P. engines (with the reduced rods, etc.—L.P. powers being usually about five-eighths of H.P. or I.P. powers) at the ends, and also permits them to take steam alternately. If the engine is a four-crank quadruple it is better to make the heavier L.P. one of the middle cylinders and keep the lighter ones at the ends.

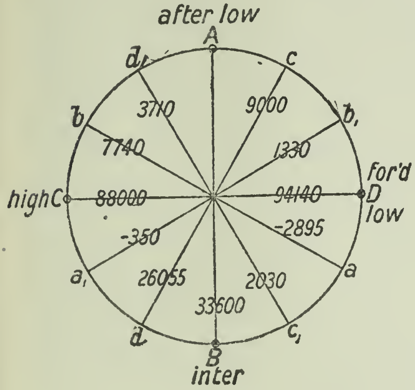

FIG. 53.

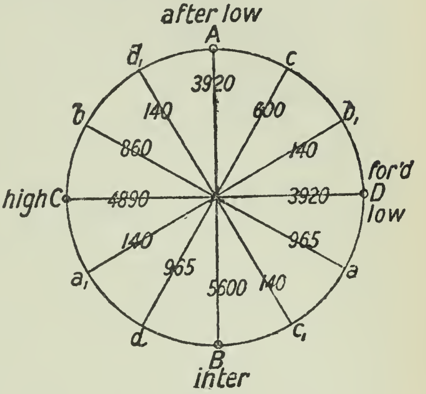

FIG. 54.

Now describe two circles (figs. 53 and 54), each representing the crank path; draw in the various cranks and eccentrics at their correct angles, and figure the moments on one and the forces on the other, as

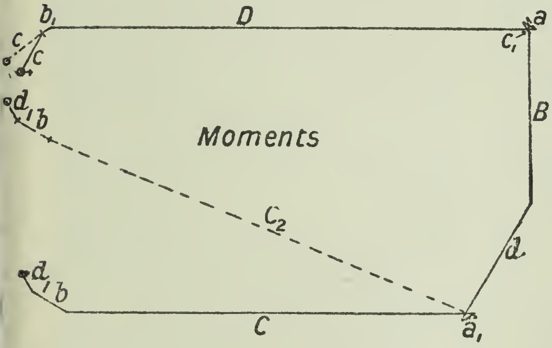

FIG. 55.

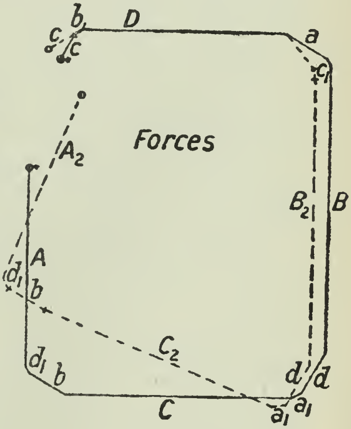

FIG. 56.

indicated. Then draw a polygon (fig. 55) in which the sides are successively parallel to the radial lines in fig. 53, and represent, to some convenient scale, the moments figured on them. Also draw a similar polygon (fig. 56) for the forces, from fig. 54).

If these polygons are closed figures, the moments and forces will be in equilibrium; but if they do not close, the additional side required to complete closure will represent, in magnitude and direction, the moment or force required to produce equilibrium. There are four principal weights, three principal distances (from centre to centre of cylinders), and four crank angles, or eleven principal variables, any one of which may be altered in moderation, and every alteration to which will have a material effect on the balance of the whole: and as the polygons (shown in full lines) do not close, it must now be decided what alterations shall be made to obtain the desired perfection of balance. As there are only three principal moments (viz., those of the piston, &c.), it is evident that the moment polygon, to close, must be roughly triangular in shape, and this shows that some alteration in crank angles will be necessary. In making such alteration it is well to bear in mind the interchangeability of the crankshafts—supposing that, as is usual, there are two of these, each carrying two cranks. To meet this requirement the angle between A and B should be the same as that between C and D; then the forward half, when reversed, may be placed in the after bearings in case of need, and *vice versâ.* The reversal is only necessary when the L.P. crank-pins are shorter than the others, as has been assumed in this case.

The dotted lines on figs. 55 and 56 show one solution of the problem. The cranks C and A are each moved 22½° ahead and the weight of the H.P. piston is increased by 410 lbs.—the figures then practically closing. It must be noted that when a crank angle is altered in these polygons the angular position of its eccentrics must also be altered; and before such an alteration is decided upon, the effect it will have should be examined both in the moment and in the force diagrams.

Similar polygons should now be constructed for the unbalanced rotating weights (which, by the way, if balanced in one plane are also in balance in any other plane through the shaft axis); and if the same crank angles, &c., do not appear to be the most suitable, some splitting of differences must be resorted to. As a rule, the weights of the rotating parts bear such a relation to those of the reciprocating parts that the same crank angles, &c., will be found to suit both.

It will, of course, be apparent that, with so many variables, more than one solution may be quite practicable.

When the leading features of a design have been worked out and settled by the methods outlined above, the results, as regards any residual unbalanced forces, may be exhibited upon a diagram such as that shown in fig. 57, which is constructed as follows:—

First, take a straight line PQ to represent the axis of the crankshaft, and lay off from it, at their proper fore and aft positions, ordinates (such as CC′, DD′, &c.) representing in magnitude and direction the total vertical force (*i.e.* force due to reciprocating parts plus vertical component of force due to rotating parts) due to each item at, say, 0° position (top centre) of high-pressure crank; and also show the vertical centre line, MM, through the middle point of the whole engine (half way between outer cranks). Then collect all the pluses and all the minuses, and subtract from the greater sum the less to find the resultant force R,—to which affix its proper sign. Then, to determine the fore and aft position of R, assume a reference plane at $XX_1$ (say), and find the moment of each force about it by multiplying the force (in lbs.) by the distance of its line of action (in feet) from $XX_1$, taking care to distinguish the products by their proper signs. And finally, subtract for resultant moment, and divide latter by resultant force R, when the dividend will be the distance, $y$, at which this force acts from $XX_1$; and a further subtraction, the position of MM being known, will give Z. In the notation used, plus moments act "clockwise," and minus moments "anti-clockwise." The conclusion is, then, that at 0° H.P. crank angle, there exists a couple of moment RZ tending to tilt up the forward end of the engines, on the after end of the bed-plate as fulcrum, and a vertical force R, acting on MM and tending to press the engines as a whole down upon their seatings.

In determining the vertical component of the centrifugal force due to the rotating parts, the angle of each crank or eccentric should be stated as the number of degrees between it and the nearest vertical radius, and will, therefore, never exceed 90°; and, using the same notation as in fig. 50, the sign round the upper 180° will be minus (upward force), and round the lower 180°, plus (downward force).

As regards the reciprocating parts, the radius of fig. 49 may be made to represent the centrifugal force due to 1000 lbs., and when the radial lines have been drawn at the correct angles and the acceleration forces (LM, &c.) determined, the latter may be multiplied by the number of thousands, or by the decimal of a thousand, requisite to give the total force in each case.

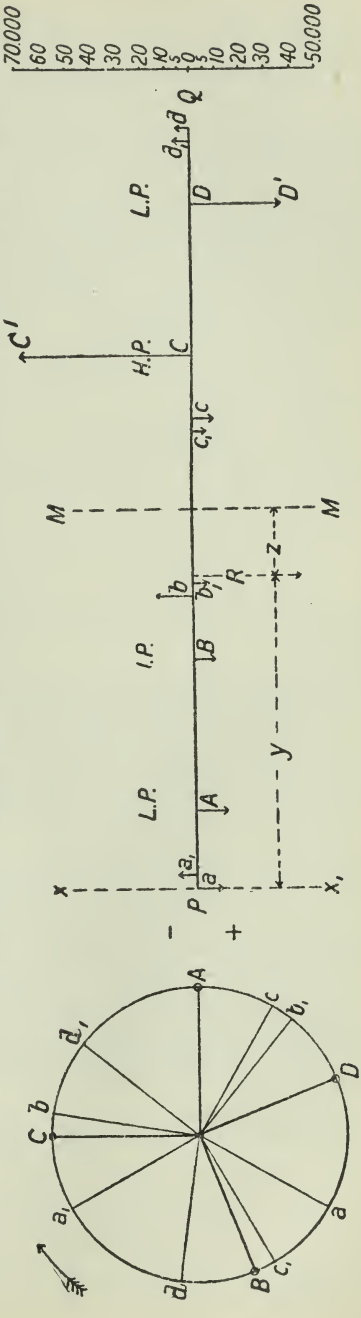

FIG. 57.

The moment RZ and the force R may now be represented by ordinates set up at 0° (fig. 58), and when similar ordinates have been determined for other crank positions (*e.g.* CD at 120°), curves may be drawn through their extremities, as in the figure, and these show at a glance the variations in the tilting moment and force during one complete revolution. It will, of course, be seen that if the engines were perfectly balanced these curves would become straight lines and would coincide with the base line; but this is a state of things rarely or never reached in practice, as, to mention one point only, estimated weights cannot be expected to tally exactly with the final weights of the parts, which have probably been altered in many points of detail since the first design—on which the balancing calculations were based—was got out.

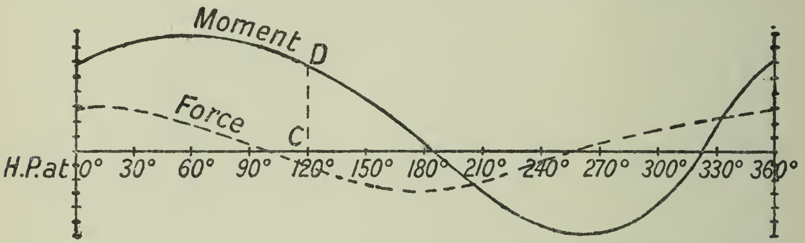

Fig. 58.

Similar curves may be constructed to show the residual moments and forces in the horizontal plane.

To give a clear idea of the condition of any set of engines as regards balance three diagrams should be presented, viz.:—

(*a*) Curves of residual vertical forces and moments.
(*b*) Curves of residual horizontal forces and moments.
(*c*) Curve of twisting moment.

And each diagram should have a scale of tons, or foot-tons, marked upon it; and the twisting moment diagram should also show the greatest and least moment as percentages of the mean. Three such diagrams were latterly required by the British Admiralty with each new design of a reciprocating engine submitted by a contractor.

Amongst other causes of vibration of minor importance may be mentioned—(*a*) the pendulum-like swing of the connecting-rod; (*b*) the thrusts on the piston-rod guides; and (*c*) irregular action of the propeller, due to disturbance of the water supply caused by the hull and fittings.

Every propeller should, as a matter of course, be accurately balanced.

# BOILERS.

## Fuels, Combustion, &c.

**Coal.**—The chief varieties of coal and their leading characteristics are as follows:—

(1) *Anthracite*, consisting almost entirely of free carbon, generally jet black in appearance, but sometimes greyish like plumbago, has a specific gravity generally of about 1·5, but sometimes as high as 1·9; it burns without emitting flame or smoke, but requires a strong draught to burn at all. It is capable of evaporating (theoretically) nearly 16 times its weight of water, but to obtain good results from it careful stoking is necessary, as when suddenly exposed to heat it does not cake but becomes very friable, breaks up into small pieces, and falls through the bar-spaces if disturbed much. The fires should be worked light when using it, and the coal carefully spread. The heat is very intense and local, so that furnaces intended to burn it should be high in the crowns and protected at the sides.

(2) *Dry bituminous coal* contains from 70 to 80 per cent. of carbon, and about 15 per cent. of volatilizable matter; its specific gravity is from 1·3 to 1·45. It burns easily and swells considerably while being converted into coke. The harder kinds do not burn so readily, nor do the pieces stick together so easily when burning, and are generally well adapted for marine boilers.

(3) *Bituminous caking coal*, containing from 50 to 60 per cent. of carbon, is generally of about the same specific gravity as the dry bituminous; it contains, however, as much as 30 per cent. of volatilizable matter, and consequently develops hydrocarbon gases; it burns with a long flame, and sticks together in caking, so as to lose all trace of the original forms of the pieces. It requires special means to prevent smoke.

(4) *Cannel coal*, or *long-flaming coal.*—This is seldom used for steam purposes, as it gives off large quantities of smoke, and is very scarce. It is the best coal for the manufacture of gas.

(5) *Lignite*, or *brown coal*, is of later formation than the other coals, and in some instances approaches to a peaty nature. It contains, however, when good, from 56 to 76 per cent. of carbon, and has a specific gravity from 1·20 to 1·35. It also contains large quantities of oxygen, and a small quantity of hydrogen. The commoner kinds of lignite are poor, and contain as little as 27 per cent. of carbon, and therefore are not suitable for steaming purposes.

**Wood.**—Dry wood contains on an average about 50 per cent. of carbon, 41 of oxygen, and 6 of hydrogen.

**Patent fuels.**—These usually consist of coal-dust, mixed with a little coal-tar and pressed into hard bricks; their value depends very much on the quality of coal from which they are made.

**The value of a fuel** is determined by its chemical composition. All fuels contain more or less carbon, most have also hydrogen and oxygen in various proportions, and some contain small quantities of nitrogen, sulphur, &c., in addition.

The ordinary symbols for these substances and their combining weights are as follows:—

| | | | | |
|---|---|---|---|---|
| Carbon | — symbol | C | — combining weight | 12. |
| Hydrogen | ,, | H | ,, | 1. |
| Oxygen | ,, | O | ,, | 16. |
| Nitrogen | ,, | N | ,, | 14. |
| Sulphur | ,, | S | ,, | 32. |

When they combine chemically one with another, as in combustion, they invariably do so in the ratio of their combining weights, or of some multiple of the same. Thus,—

Carbonic oxide — symbol $CO = C_{12} + O_{16}$.
Carbonic dioxide ,, $CO_2 = C_{12} + O_{32}$.
Water . . ,, $H_2O = H_2 + O_{16}$.
Olefiant Gas. ,, $CH_2 = C_{12} + H_2$.
Marsh gas . ,, $CH_4 = C_{12} + H_4$.

Atmospheric air when dry is a mechanical mixture of nitrogen and oxygen in the proportion of 77 parts by weight of the former to 23 of the latter.

The heat developed by any substance during combustion is called its *total heat of combustion*, and is measured in *units of heat* or *thermal units.*

**The British thermal unit** is the amount of heat required to raise the temperature of one pound of pure water one degree Fahrenheit when at or near its greatest density (39·1° F.).

**The mechanical equivalent of heat** is the number of foot-pounds of energy required to raise the temperature of one pound of water one degree Fahrenheit; or in other words,—

1 British thermal unit* = 778 foot-pounds,
or 1 Horse-power = 33,000 foot-pounds = 42·42 thermal units.

To burn fully 1 lb. of carbon or convert it into carbon dioxide ("carbonic acid"), 2·7 lbs. of oxygen, or 12 lbs. of air are required, and a total heat of combustion of 14,500 units is developed,—*i.e.* sufficient to evaporate 15 lbs. of water from and at 212°. If, however, the air supply is restricted or deficient, there is a tendency to the production of carbon monoxide or "carbonic oxide,"—a product requiring only half the amount of oxygen for its formation,—and the total heat developed is then only 4400 units.

Each pound of hydrogen requires 8 lbs. of oxygen or 36 lbs. of air for its complete combustion, and develops in combining 62,032 units of heat,—sufficient to evaporate 64 lbs. of water from and at 212°.

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* Joule's *mechanical equivalent*, 772, was for many years accepted as B.T. unit, but later experiments of a more refined nature have proved it to be 778 ft.-lbs.

Sulphur exists only in small quantities in good coal, and its total heat of combustion is only about 4000 units per pound.

When two substances combine chemically, as in combustion, the weight of the products of combustion is the sum of the weights before combination,—*e.g.*, 1 lb. of carbon uniting with 12 lbs. of air gives 3·7 lbs. of carbonic acid diluted by 9·3 lbs. of nitrogen; and the temperature of the products is found by dividing the total heat of combustion by their weight multiplied by their specific heat.

**The specific heat** of any substance is the quantity of heat required to raise the temperature of a pound of it one degree Fahrenheit, and is measured by the ratio that this quantity bears to the quantity required to raise the temperature of a pound of water one degree,—*i.e.* to the British thermal unit; thus, the specific heat of hydrogen, at a pressure of 30 inches of mercury, is 3·4, that of carbonic acid ·216.

When oxygen and hydrogen are both present in a fuel they are in combination as water and become steam without developing any heat; it is therefore only when there is an *excess* of hydrogen that any heating effect is produced. The only effect of nitrogen is to reduce the intensity of combustion and lower temperature of products.

The following rules are deduced from the above considerations:—

**Rule 238.** Total heat of combustion of one pound of fuel,—in thermal units. $\Big\} = 14{,}500\left\{C + 4\cdot28\left(H - \frac{O}{8}\right)\right\}$.

**Rule 239.** Theoretical evaporative power of one pound of fuel,—expressed in pounds of water. $\Big\} = 15\left\{C + 4\cdot28\left(H - \frac{O}{8}\right)\right\}$.*

**Rule 240.** Number of pounds of air required to burn one pound of fuel. $\Big\} = 12C + 36\left(H - \frac{O}{8}\right)$.

Ordinary coal or coke requires for its complete combustion 12 lbs. of air per pound of the fuel, but in practice, with chimney draught, twice this quantity may be supplied, as much passes free through the furnace in an uncombined state. Kennedy found that 18 lbs. of air gave the best results.

At the temperature of 62° F. and at the sea-level, the volume of 1 lb. of air is 13·14 cubic feet; therefore 315 cubic feet of air are necessary for the proper combustion of 1 lb. of coal or coke in an ordinary furnace. If artificial or forced draught is employed, this quantity may be reduced to about 250 cubic feet, more or less, according to the force of the draught.

The following Table gives additional information as to the composition, total heat of combustion, and evaporative value of various fuels:—

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* Allowing for the 212° below which there can be no abstraction of heat for evaporation, the multiplier should be 14·78 instead of 15.

## Table LXXXVII.—Composition and Value of Fuels.

| Description. | Carbon. | Hydrogen. | Oxygen. | Sulphur. | Ash, &c. including Nitrogen. | Total Heat of Combustion. | Evaporation from and at 212° or B.T.U.÷966° | One ton occupies in Cubic Feet. |
|---|---|---|---|---|---|---|---|---|
| | | | | | | Units. | Pounds. | |
| Welsh—Ebbw Vale, | 87·78 | 5·15 | 0·39 | 1·02 | 3·66 | 16221 | 16·79 | ... |
| ,, Powell's Duffryn, | 88·26 | 4·66 | 0·60 | 1·77 | 4·71 | 15788 | 16·34 | ... |
| ,, Llangennech, | 34·97 | 4·26 | 3·50 | 0·42 | 6·85 | 14682 | 15·20 | ... |
| ,, Graigola, | 84·87 | 3·84 | 7·19 | 0·45 | 1·91 | 14130 | 14·63 | ... |
| ,, Average, | 83·87 | 4·79 | 4·15 | 1·43 | 5·89 | 14858 | 15·52 | 42·7 |
| Newcastle Average, | 82·12 | 5·31 | 5·69 | 1·24 | 5·12 | 14820 | 15·32 | 45·3 |
| Derbyshire Average, | 79·68 | 4·94 | 10·28 | 1·01 | 4·06 | 13860 | 14·34 | 47·4 |
| South Yorkshire, | 81·88 | 4·83 | 7·47 | 0·54 | 2·95 | 14296 | 14·71 | 46·0 |
| Lancashire Average, | 77·90 | 5·32 | 9·53 | 1·44 | 6·18 | 13918 | 14·56 | 45·2 |
| Scotch ,, | 78·53 | 5·61 | 9·69 | 1·11 | 5·03 | 14164 | 14·65 | 42·0 |
| Irish Anthracite, | 80·03 | 2·30 | ... | 6·76 | 11·03 | 13302 | 14·5 | 35·7 |
| American Anthracite, | 88·54 | 2·43 | ... | 0·04 | 8·60 | 15163 | 15·67 | 42·35 |
| ,, Bituminous, | 73·21 | ... | ... | 0·42 | 11·27 | ... | ... | 42·44 |
| French Anthracite, | 86·17 | 2·67 | 2·85 | ... | 8·56 | 14038 | 14·53 | 40·00 |
| ,, Hard Bituminous, | 88·56 | 4·88 | 4·38 | ... | 2·19 | 15525 | 16·10 | 42·75 |
| ,, Caking ,, | 87·73 | 5·08 | 5·65 | ... | 1·54 | 15422 | 16·00 | 42·75 |
| Chilian Coal, | 63·56 | 5·43 | 14·84 | 2·50 | 14·13 | 11030 | 11·68 | ... |
| Indian ,, Average, | 70·20 | ... | ... | ... | 22·9 | ... | ... | ... |
| Patent Fuel—Warlichs, | 90·02 | 5·56 | ... | 1·62 | ... | 16495 | 17·07 | ... |
| ,, ,, Average, | 83·40 | 4·97 | 2·79 | 1·26 | 6·01 | 15000 | 15·66 | 34·4 |
| Natal Coal—Navigation dry | 81·38 | 5·45 | 2·82 | 1·06 | 9·29 | 14045 | 14·54 | ... |
| Kentish Coal | 79·78 | ... | 16·73 | 0·8 | 2·08 | 13900 | 14·4 | ... |
| Coke—Best Durham, | 85 to 92 | ... | ... | 0·25 to 2·0 | 4 to 12 | 12832 | 13·30 | ... |
| Woods—Oak, | 49·64 | 5·92 | 41·16 | ... | 3·26 | average | | 94·4 |
| ,, Birch, | 50·20 | 6·20 | 41·62 | ... | 1·96 | when dry | | 106·1 |
| ,, Fir, | 51·79 | 6·28 | 41·93 | ... | ... | 7800 | 8·1 | ... |
| Peat—Fairly dry, | 59·6 | 5·80 | 29·6 | 0·3 | 4·7 | 9951 | 10·30 | ... |
| Petrol'm—Americ'n crude | 84·7 | 13·1 | 2·2 | ... | ... | 20240 | 20·33 | 39 to 46 |
| ,, Refuse, | 87·1 | 11·7 | 1·2 | ... | ... | 19832 | 20·53 | 40 |
| ,, Caucasian crude, | 86·6 | 12·3 | 1·1 | ... | ... | 20138 | 20·85 | 42 |

## Table LXXXVIIa.—Liquid Fuels. Composition and Heating Value of Oils (*from Redwood*).

| | Specific Gravity at 32° F. | Chemical Composition. | | | Actual evaporation per lb. of Fuel from 212° F. | British Thermal Units per lb. |
|---|---|---|---|---|---|---|
| | | Carbon. | Hydrogen. | Oxygen. | | |
| | | Per cent. | Per cent. | Per cent. | Lbs. | |
| Heavy petroleum from West Virginia | 0·873 | 83·5 | 13·3 | 3·2 | 14·58 | 18,324 |
| Light petroleum from West Virginia | 0·841 | 84·3 | 14·1 | 1·6 | 14·55 | 18,401 |
| Light petroleum from Pennsylvania | 0·816 | 82·0 | 14·8 | 3·2 | 14·05 | 17,933 |
| Heavy petroleum from Pennsylvania | 0·886 | 84·9 | 13·7 | 1·0 | 15·30 | 19,210 |
| American petroleum | 0·820 | 83·4 | 14·7 | 1·9 | 14·14 | 17,588 |
| Petroleum from Parma | 0·786 | 84·0 | 13·4 | 1·8 | 13·96 | 18,218 |
| ,, Pechelbronn | 0·912 | 86·9 | 11·8 | 1·3 | 14·30 | 17,474 |
| ,, ,, | 0·892 | 85·7 | 12·0 | 2·3 | 14·48 | 18,036 |
| ,, Schwabweiler | 0·861 | 86·2 | 13·3 | 0·5 | 15·36 | 18,824 |
| ,, East Galicia | 0·870 | 82·2 | 12·1 | 5·7 | 14·23 | 18,153 |
| ,, West Galicia | 0·885 | 85·3 | 12·6 | 2·1 (NO) | 14·79 | 18,416 |
| Shale oil from Ardeche, France | 0·911 | 80·3 | 11·5 | 8·2 (0·5 N) | 12·24 | 16,283 |
| Coal tar from Paris Gas Works | 1·044 | 82·0 | 7·6 | 10·4 | 12·77 | 16,049 |
| Petroleum from Balakhany | 0·882 | 87·4 | 12·5 | 0·1 | 17·41 | 21,060 |
| Light petroleum from Baku | 0·884 | 86·3 | 13·6 | 0·1 | 16·40 | 20,628 |
| Heavy ,, ,, | 0·938 | 86·6 | 12·3 | 1·1 | 15·55 | 19,440 |
| Petroleum residues from the Baku factories | 0·928 | 87·1 | 11·7 | 1·2 | 15·54 | 19,260 |
| Petroleum from Java | 0·923 | 87·1 | 12·0 | 0·9 | 15·02 | 19,496 |
| Heavy oil from Ogaio | 0·985 | 87·1 | 10·4 | 2·5 | 14·75 | 18,146 |
| ,, ,, ,, Mexico, crude | 0·950 | 83·5 | 11·7 | 1·37 (NO) | ... | 18,900 |
| ,, ,, ,, California, crude | 0·966 | 81·5 | 11·6 | 6·92 (NO) | ... | 18,670 |
| ,, ,, ,, Texas, crude | 0·924 | 84·6 | 10·9 | 2·57 | ... | 18,800 |

## Table LXXXVIIb.—Consumption of Various Oil Fuels in Gallons per Twenty-four Hours.

*N.B.*—For Scotch Shale, multiply by 0·948; for Good Crude, Specific Gravity (0·9), multiply by 0·905.

| B.H.P. | Consumption of Paraffin, Specific Gravity 0·815, in fractions of lbs. per B.H.P. hour. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0·300 | 0·325 | 0·350 | 0·375 | 0·400 | 0·425 | 0·450 | 0·475 | 0·500 | 0·525 | 0·550 | 0·575 | 0·600 | 0·650 |
| 50 | 44·18 | 47·86 | 51·54 | 55·22 | 58·90 | 62·58 | 66·26 | 69·94 | 73·62 | 77·30 | 80·98 | 84·66 | 88·34 | 95·7 |
| 75 | 66·26 | 71·78 | 77·30 | 82·83 | 88·35 | 93·87 | 99·39 | 104·9 | 110·4 | 116·0 | 121·5 | 127·0 | 132·5 | 143·6 |
| 100 | 88·35 | 95·71 | 103·6 | 110·4 | 117·8 | 125·2 | 132·5 | 139·9 | 147·2 | 154·6 | 162·0 | 169·3 | 176·7 | 191·4 |
| 125 | 110·4 | 119·6 | 128·8 | 138·1 | 147·3 | 156·5 | 165·7 | 174·9 | 184·1 | 193·3 | 202·5 | 211·7 | 220·9 | 239·3 |
| 150 | 132·5 | 143·5 | 154·6 | 165·7 | 176·7 | 187·7 | 198·8 | 209·8 | 220·9 | 231·9 | 243·0 | 253·1 | 265·0 | 287·1 |
| 175 | 154·6 | 167·5 | 180·4 | 193·3 | 206·2 | 219·0 | 231·9 | 244·8 | 257·7 | 270·6 | 283·4 | 296·3 | 309·2 | 335·0 |
| 200 | 176·7 | 191·4 | 206·2 | 220·9 | 235·6 | 250·3 | 265·0 | 278·8 | 294·5 | 309·2 | 323·9 | 338·6 | 353·4 | 382·8 |
| 225 | 198·8 | 215·4 | 232·0 | 248·5 | 265·1 | 281·6 | 298·2 | 314-7 | 331·3 | 347·9 | 364·4 | 381·0 | 397·5 | 430·7 |
| 250 | 220·9 | 239·3 | 257·7 | 276·1 | 294·5 | 312·9 | 331·3 | 349·7 | 368·1 | 386·5 | 404·9 | 423·3 | 441·7 | 478·5 |
| 275 | 243·0 | 263·3 | 283·5 | 303·7 | 324·0 | 344·2 | 364·4 | 384·7 | 404·9 | 425·2 | 445·4 | 465·3 | 485·9 | 526·4 |
| 300 | 265·1 | 287·2 | 309·3 | 331·3 | 353·4 | 375·5 | 397·6 | 419·7 | 441·7 | 463·8 | 485·9 | 507·7 | 529·9 | 574·2 |
| 325 | 287·1 | 311·0 | 334·9 | 358·9 | 332·9 | 406·8 | 430·7 | 454·6 | 478·5 | 502·5 | 526·4 | 550·0 | 573·2 | 622·1 |
| 350 | 309·2 | 335·0 | 360·7 | 386·5 | 412·3 | 438·1 | 463·8 | 489·6 | 515·3 | 541·1 | 566·9 | 592·3 | 618·4 | 669·9 |
| 400 | 353·4 | 382·9 | 412·3 | 441·7 | 471·2 | 500·6 | 530·1 | 559·5 | 588·9 | 618·4 | 647·8 | 677·0 | 706·7 | 765·6 |
| 450 | 397·6 | 430·7 | 463·9 | 496·9 | 530·1 | 563·2 | 596·3 | 629·5 | 662·6 | 695·7 | 728·8 | 761·6 | 795·1 | 861·3 |
| 500 | 441·8 | 478·6 | 515·4 | 552·2 | 589·0 | 625·8 | 662·6 | 699·4 | 736·2 | 773·0 | 809·8 | 846·6 | 883·4 | 957·0 |
| 550 | 485·9 | 526·4 | 566·9 | 607·4 | 647·9 | 688·4 | 728·9 | 769·4 | 809·8 | 850·3 | 890·8 | 931·3 | 971·2 | 1053 |
| 600 | 530·1 | 574·2 | 618·4 | 662·3 | 706·8 | 751·0 | 795·1 | 839·3 | 883·4 | 927·6 | 971·8 | 1016 | 1060 | 1147 |
| 650 | 574·3 | 622·2 | 670·0 | 717·8 | 765·7 | 813·5 | 861·4 | 909·2 | 957·1 | 1005 | 1053 | 1101 | 1148 | 1243 |
| 700 | 618·5 | 670·1 | 721·6 | 773·0 | 824·6 | 876·1 | 927·6 | 979·2 | 1031 | 1082 | 1134 | 1185 | 1237 | 1339 |
| 750 | 662·6 | 717·8 | 773·0 | 828·3 | 883·5 | 938·7 | 993·9 | 1049 | 1104 | 1160 | 1215 | 1270 | 1325 | 1435 |
| 800 | 706·8 | 765·7 | 824·6 | 883·5 | 942·4 | 1001 | 1060 | 1119 | 1178 | 1237 | 1296 | 1355 | 1413 | 1530 |
| 850 | 751·0 | 813·6 | 876·2 | 938·7 | 1001 | 1064 | 1126 | 1189 | 1252 | 1314 | 1377 | 1439 | 1502 | 1626 |
| 900 | 795·1 | 861·4 | 927·5 | 993·9 | 1060 | 1126 | 1192 | 1259 | 1325 | 1391 | 1458 | 1524 | 1590 | 1722 |
| 950 | 839·3 | 909·3 | 979·0 | 1049 | 1119 | 1189 | 1258 | 1329 | 1399 | 1468 | 1530 | 1609 | 1678 | 1818 |
| 1000 | 883·5 | 957·1 | 1031 | 1104 | 1178 | 1251 | 1325 | 1399 | 1472 | 1546 | 1620 | 1693 | 1767 | 1913 |

## Table LXXXVIIc.—Liquid Fuels as in Practice.

| Fuel. | Specific Gravity. | Ash, per Cent. | Water, per Cent. | Sulphur, per Cent. | Gross Heat Value. | | Nett Heat Value. | |
|---|---|---|---|---|---|---|---|---|
| | | | | | Per Lb. | Per Gallon. | Per Lb. | Per Gallon. |
| | | | | | B.T.U. | B.T.U. | B.T.U. | B.T.U. |
| British Navy oil | 0·907 | 0·023 | 0·500 | 0·750 * | 19,250 | 174,598 | 17,900 | 162,353 |
| Roumanian gas oil | 0·871 | ... | trace | ... | 19,280 | 167,929 | 18,010 | 156,867 |
| ,, ordinary oil | 0·928 | 0·051 | ,, | 0·430 | 18,920 | 175,578 | 17,685 | 164,117 |
| ,, residues | 0·930 | 0·133 | ,, | ... | 19,010 | 176,793 | 17,730 | 163,889 |
| Trinidad crude oil | 0·964 | 0·141 | ,, | 3·360 | 18,400 | 177,376 | 17,230 | 166,097 |
| Tarakan ,, | 0·948 | 0·035 | ,, | 0·250 | 18,870 | 178,822 | 17,650 | 161,322 |
| Californian ,, | 0·945 | ... | ... | ... | 18,080 | 170,856 | 16,900 | 159,705 |
| Mexican ,, | 0·950 | ... | ... | 3·270 | 18,800 | 178,600 | 17,580 | 167,010 |
| Texas ,, | 0·924 | ... | ... | 1·63 | 19,000 | 175,560 | 17,770 | 164,195 |
| Texas and Californian residuals | ... | ... | ... | 0·90 | 19,200 | ... | 18,000 | ... |

* The Admiralty now permit up to 3 per cent of Sulphur in Oil Fuel for steam boilers, *v.* p. 308.

Table LXXXVIId.—Oil Fuels as Used in U.S. American Navy Boilers.

| | Crude Texas Oil. | Texas Oil as Used. | Californian Oil as Used. | Residuum of Mixed Texas and Californian. | Coal, Pocahontas Handpicked and Screened. |
|---|---|---|---|---|---|
| Carbon, content per cent. | 84·60 | 83·26 | 81·52 | 84·35 | 85·94 |
| Hydrogen ,, ,, | 10·90 | 12·41 | 11·01 | 11·33 | 4·45 |
| Oxygen ,, ,, | 2·87 | 3·83 | 6·92 | 2·82 | 4·50 |
| Nitrogen ,, ,, | ... | ... | | 0·60 | 1·14 |
| Sulphur ,, ,, | 1·63 | 0·50 | 0·55 | 0·90 | 0·82 |
| Specific gravity at 60° F. | 0·924 | 0·926 | 0·966 | 0·966 | ... |
| Flash point . . F.° | 180 | 216 | 311 | 270 | ... |
| Fire ,, . . ,, | 200 | 240 | ... | 280 | ... |
| Vaporisation point ,, | ... | 142 | 230 | ... | ... |
| Loss for 6 hours at 212° F. | ... | 21·65% | 12·20% | 14·90% | ... |
| Calorific value as calculated . . . . | 18,812 | 19,480 | 18,192 | 19,049 | 14,992 |
| Calorific value by calorimeter . . . | 19,060 | ... | 18,667 | 19,215 | ... |
| Water evaporated from and at 212° F. in a Hohenstein w.t. boiler | ... | 12·654 | 11·574 | 11·515 | 9·215 |
| Water evaporated from and at 212° F. in a Hohenstein w.t. boiler best results with air pulverisers . . | ... | 12·850 | ... | 12·660 | ... |

Table LXXXVIIe.—Liquid Fuels, Viscosity of, at various Temperatures, by Number of Seconds taken for the Passing of 12·2 cubic inches (200 c.cm.).

| Fuel. | Specific Gravity. | Temperature, F. | | | | | |
|---|---|---|---|---|---|---|---|
| | | 70°. | 85°. | 105°. | 120°. | 140°. | 160°. |
| British Navy oil . | 0·907 | 818 | 435 | 255 | 179 | 140 | 110 |
| Roumanian residues | 0·946 | ... | ... | ... | 785 | 474 | 293 |
| Trinidad crude oil . | 0·964 | ... | ... | ... | ... | ... | 1127 |
| Tarakan ,, . | 0·948 | 745 | 431 | 263 | 170 | 127 | 99 |
| Mexican ,, . | 0·950 | ... | ... | ... | ... | ... | ... |

## Rules for the Burning and Carrying of Oil Fuel.

1. In vessels fitted for burning oil fuel, the following records will be made in the Register book:—"Fitted for oil fuel F.P. above 150° F." in cases in which approval has been given for the use of high flash point oil only; and "Fitted for low flash oil fuel" in cases in which the approval covers the use of low flash point.

2. The following arrangements are applicable only to the case of oil fuel, the flash point of which (by Abel's close test) is not below 150° F. For a lower flash point the arrangements must be submitted for consideration.

3. Oil fuel, the flash point of which does not fall below 150° F., may be carried in ordinary cellular double bottoms under engines or boilers or ordinary cargo holds, in peak or in deep tanks, or in oil bunkers specially constructed.

4. Cellular double bottoms when fitted for oil fuel are to have oil-tight centre line divisions, and the lengths of these compartments are to be submitted for approval.

5. Peak tanks, deep tanks, bunkers specially constructed for oil fuel, and settling and other service tanks must be fitted with bulkhead subdivisions or wash plates to the Committee's satisfaction and strengthened to efficiently withstand the stresses on them when only partly filled and in a seaway. The riveting of these spaces is to be as required by the Rules in the cases of vessels carrying petroleum in bulk, and the scantlings and arrangements to the Committee's satisfaction.

6. All compartments intended for carrying oil fuel must be tested by a head of water extending to the highest point of the filling pipes, or 12 feet above the load-line or 12 feet above the highest point of the compartment, whichever is the greatest.

7. Each compartment must be fitted with an air-pipe, to be always open, discharging above the upper deck. All double-bottom compartments used for oil fuel should have suitable holes and doors of approved design fitted in the outer bottom plating.

8. The pumping arrangements of the oil-fuel compartments must be absolutely distinct from those of other parts of the vessel and submitted for approval. All oil-fuel suction pipes and all bilge or ballast pipes passing through oil-fuel tanks or oil bunkers are to be of iron or steel.

9. If it is intended to carry sometimes oil-fuel and sometimes water ballast in any of the compartments, the valves or cocks connecting the suction pipes to these compartments with the ballast donkey pump and those controlling them with the oil-fuel pump must be so arranged that oil may be pumped from any one compartment by the oil-fuel pump at the same time as the ballast donkey is being used on any other compartment.

10. All oil-fuel suction pipes should have valves or cocks fitted at the bulkheads where they enter the machinery space, capable of being worked both from this space and from the deck, outside of the fidely and engine-room casings. Valves or cocks similarly worked are to be fitted to all pipes leading from the settling or other tanks.

11. Oil-fuel pipes should, where practicable, be placed above the stokehold and engine-room plates, and where they are always visible.

12. No wood fittings or bearers are to be fitted in the stokehold spaces.

13. Where oil-fuel compartments are at the sides of, or above, or below the boilers, special insulation is to be fitted where necessary to protect them from the heat from the boilers, smoke boxes, casings, &c.

14. Water service pipes and hoses are to be fitted so that the stokehold plates can at any time be flushed with sea water into the bilges.

15. If the oil fuel is sprayed by steam, means are to be provided to make up for the fresh water used for this purpose.

16. If the oil fuel is heated by a steam coil, the condensed water should not be taken directly to the condensers, but should be led into a tank or an open funnel mouth, and thence led to the hot well or feed tank.

## Admiralty Conditions of Contract for Oil Fuel.

1. *Quality.*—The oil fuel supplied under this contract shall consist of liquid hydrocarbons, and may be either:—

(*a*) Shale oil; or
(*b*) Petroleum as may be required; or
(*c*) A distillate or a residual product of petroleum;

and shall comply with the Admiralty requirements as regards flash point, fluidity at low temperatures, percentage of sulphur, presence of water, acidity, and freedom from impurities.

The flash point shall not be lower than 175° F., close test (Abel or Pensky-Martens).*

The proportion of sulphur contained in the oil shall not exceed 3 per cent.

The oil fuel supplied shall be as free as possible from acid, and in any case the quantity of acid must not exceed 0·5 per cent., calculated as oleic acid when tested by shaking up the oil with distilled water, and determining by titration with decinormal alkali the amount of acid extracted by the water, methyl orange being used as indicator.

The quantity of water delivered with the oil shall not exceed 0·5 per cent.

The viscosity of the oil supplied shall not exceed 2000 seconds for an outflow of 50 cubic centimetres at a temperature of 32° F., as determined by Sir Boverton Redwood's standard viscometer (Admiralty type for testing oil fuel).

The oil supplied shall be free from earthy, carbonaceous, or fibrous matter, or other impurities which are likely to choke the burners.

The oil shall, if required by the inspecting officer, be strained by being pumped on discharge from the tanks, or tank steamer, through filters of wire gauze having 16 meshes to the inch.

The quality and kind of oil supplied shall be fully described. The original source from which the oil has been obtained shall be stated in detail, as well as the treatment to which it has been subjected, and the place at which it has been treated.

The ratio which the oil supplied bears to the original crude oil should also be stated as a percentage.

**Rate of combustion.**—In the mercantile marine, when working economically with chimney draught only, the coal burned per square foot of grate per hour varies from about 15 lbs., with long bars (6 feet to 6 feet 6 inches), to about 20 lbs. when the length of the bars does not exceed 1·33 × diameter of furnace, and all calculations for grate areas of merchant steamers should be based on these figures; for, although larger quantities may be *consumed* by forcing the fires, or may, perhaps, be completely burned when the wind is strong and draught good, there will be, on the other hand, many days when the quantity burned will be less than the average given above.

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* In the case of oils of exceptionally low viscosity, such as distillates from shale, the flash point must be not less than 200° F.

With forced or artificial draught much larger quantities are consumed: with the Howden system as much as 40 lbs. can be economically burnt, and even up to 60 lbs. with somewhat less economic results.

The higher efficiency of the shorter bars is largely due to the better air supply, since, with a given diameter of furnace, the area at the mouth of the ash-pit is the same whether the bars are long or short; and it is no doubt for this reason that the Macfarlane Gray rule—that *the consumption of coal per foot of grate is very nearly proportional to the diameter of the furnace*—is found to be so nearly correct in everyday practice. Another cause of the higher efficiency of short bars is the fact that bars over about 5 feet in length cannot be so well stoked by hand, with the result that the fire burns into holes at the back end, and allows cold air to rush in,—thus depriving the other parts of the fire of their due proportion of air and reducing the efficiency of the furnace. In practice, with average stokers, the efficiency of a grate over 4 feet in length is nearly inversely as its length.

The rate of combustion also naturally depends a good deal on the quality or class of coal burned, but by far the most important factor is the strength of the draught.

**Chimney or funnel.**—The draught obtained with any chimney depends mainly on its sectional area, its height, and the difference of absolute temperature between the gases in it and the external air.

Professor Rankine gave the following formulæ for chimney draught:—

Let $w$ be the weight of fuel burned in a given furnace per second in pounds.

$V_0$ the volume at 32° of the air supplied per pound of fuel.

$\tau_0$ the *absolute* temperature at 32° Fahr., which is $461° + 32°$.

$\tau_1$ the absolute temperature of the gas discharged by the chimney, whose sectional area is A: then,—

**Rule 241.** Velocity of the current in the chimney in feet per second is

$$= \frac{w \times V_0 \times \tau_1}{A \times \tau_0}.$$

The density of that current in pounds to the cubic foot is very nearly

$$= \frac{\tau_0}{\tau_1}\left(0{\cdot}0807 + \frac{1}{V_0}\right);$$

that is to say, from 0·084 to $0{\cdot}987 \times (\tau_0 \div \tau_1)$.

Let $l$ denote the whole length of the chimney, and of the flue leading to it, in feet;

$m$ its "hydraulic mean depth"; that is, its area divided by its perimeter; which, for a square or round flue or chimney, is one quarter of the diameter or side;

$f$, a co-efficient of friction, whose value for currents of gas moving over sooty surfaces is estimated by Peclet at 0·012;

G, a factor of resistance for a passage of air through the grate and the layer of fuel above it, whose value, according to the experiments of Peclet on furnaces burning from 20 to 24 pounds of coal per square foot of grate per hour, is 12.

Then, according to Peclet's formula,—

**Rule 242.** The *head* required to produce the draught in question is

$$=\frac{\mu^2}{2g}\left(1+\mathrm{G}+\frac{f.l}{m}\right),$$

which, with the values assigned by Peclet to the constants, becomes

$$=\frac{\mu^2}{2g}\left(13+\frac{0\cdot012\times l}{m}\right).$$

When the *head* is given the value of $\mu$ may be calculated, and then,

**Rule 243.** Weight of fuel which the furnace is capable of burning *completely* per hour

$$=\frac{\mu\times\mathrm{A}\times\tau_0}{\mathrm{V}_0\times\tau_1}.$$

It is usual to reckon the *head* by taking one inch of water as the unit; then,—

$$\text{Head in inches of water}=0\cdot192\times h\times\frac{\tau_0}{\tau_1}\left(0\cdot0807+\frac{1}{\mathrm{V}_0}\right),$$

Sir J. Thornycroft found, by careful experiment with loco. boilers in torpedo-boats working with a *plenum* (that is, with a closed stokehole into which air is forced), "that of the initial pressure, the resistance of the tubes accounts for about seven-tenths of the whole, the resistance of the fire and fire-bars being only about one-tenth;" and that "the pressure in the funnel, as measured, was sensibly equal to atmospheric pressure."

Professor Rankine also stated that if H be the height of the funnel, $\tau_2$ the absolute temperature of the external air, then,—

**Rule 244.** Head produced by chimney draught $=\mathrm{H}\left(0\cdot96\frac{\tau_1}{\tau_2}-1\right)$,

or, taking $h$ as the head,

**Rule 244a.** Height of chimney required to produce a given draught

$$=h\div\left(0\cdot96\frac{\tau_1}{\tau_2}-1\right).$$

The velocity of the gas in the chimney is proportional to $\sqrt{h}$, and therefore to $\sqrt{0\cdot96\tau_1-\tau_2}$.

The density of that gas is proportional to $\frac{1}{\tau_1}$.

The weight discharged per second is proportional to velocity × density, and therefore, to $\frac{\sqrt{0{\cdot}96\tau_1 - \tau_2}}{\tau_1}$; which expression becomes a maximum, when $\tau_1 = \frac{25}{12}\tau_2$. Therefore the best chimney draught takes place when the absolute temperature of the gas in the chimney is to that of the external air as 25 to 12.

When this condition is fulfilled $h = H$.

That is, the height of the chimney for the best draught is equal to the *head* expressed in hot gas, and the density of the hot gas is half that of the air.

From the above it appears that the best temperature of gases at the base of the funnel is about 600°, and that therefore nearly one-fourth of the total heat of combustion is absorbed in creating the draught. The temperature of steam at 250 lbs. pressure absolute is 400° F. ; it is therefore necessary, in order to maintain the efficiency of the heating surface, to have a temperature at funnel base in excess of this, and when there is a superheater requiring a temperature of 550° to 600° F., it is obvious that the hot gases when leaving it will be above those, so that with modern conditions 600° at the funnel base is really low, and it is not uncommon to have 700° to 750° there with the proportions of heating surface to grate surface, size of tubes, &c., commonly employed ; but lower temperatures can easily be reached by using artificial draught and small tubes (say 2½ inch) fitted with retarders.

**The diameter and height of funnel** are often fixed by arbitrary methods rather than rules. The following, however, are simple and easily handled.

**Rule 244b. Height of chimney in feet** $= {\cdot}007\left(\frac{C}{A}\right)^2$.

**Rule 244c. Area of chimney in square feet** $= \frac{C \times {\cdot}084}{\sqrt{H}}$.

Where H = Height in feet.
A = Area of section in square feet.
C = Consumption in lbs. per hour on grates.

The following is a rough rule for modern practice :—

**Rule 244d. Diameter of chimney in inches** $= \sqrt{\text{I.H.P} \times F}$.

| | | |
|---|---|---|
| For Ordinary Merchant Steamers | F = 2·38 | Natural Draft. |
| ,, Naval and Express Steamers | F = 2·16 | Assisted ,, |
| ,, ,, ,, ,, | F = 1·66 | Forced ,, |

When the draught depends on the chimney alone, 40 feet should be regarded as the minimum height, where possible ; as the size of the vessel increases it is, of course, possible to get longer chimneys, and, in consequence, greater draught pressures, without fans. In the "Olympic" the tops of the chimneys are at 160 feet above the dead plates.

## Table LXXXVIII.—Capacity of Funnels for quantities of Fuel burnt in lbs. per hour.

| Diameter of Funnel. | Height of Funnel above dead plates in feet. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 20 | 25 | 30 | 35 | 40 | 50 | 60 | 70 | 80 | 90 | 100 | 110 | 120 | 130 | 140 |
| Feet. | | | | | | | | | | | | | | | |
| 2·00 | 167 | 187 | 207 | 224 | 239 | 264 | 288 | 311 | 333 | .. | .. | .. | .. | .. | .. |
| 2·25 | 213 | 239 | 261 | 282 | 303 | 337 | 370 | 399 | 426 | .. | .. | .. | .. | .. | .. |
| 2·50 | 261 | 292 | 320 | 346 | 370 | 413 | 453 | 489 | 522 | .. | .. | .. | .. | .. | .. |
| 2·75 | 316 | 354 | 387 | 419 | 448 | 500 | 548 | 591 | 632 | .. | .. | .. | .. | .. | .. |
| 3·00 | .. | 421 | 464 | 500 | 535 | 595 | 653 | 705 | 753 | 799 | .. | .. | .. | .. | .. |
| 3·25 | .. | 495 | 543 | 586 | 627 | 700 | 767 | 829 | 885 | 940 | .. | .. | .. | .. | .. |
| 3·50 | .. | 575 | 630 | 681 | 728 | 813 | 891 | 963 | 1,028 | 1,091 | .. | .. | .. | .. | .. |
| 3·75 | .. | 660 | 723 | 780 | 834 | 932 | 1022 | 1,103 | 1,180 | 1,251 | .. | .. | .. | .. | .. |
| 4·00 | .. | 750 | 822 | 888 | 948 | 1061 | 1161 | 1,254 | 1,341 | 1,422 | 1,496 | .. | .. | .. | .. |
| 4·25 | .. | .. | 926 | 1000 | 1070 | 1195 | 1310 | 1,415 | 1,510 | 1,604 | 1,692 | .. | .. | .. | .. |
| 4·50 | .. | .. | 1040 | 1123 | 1291 | 1342 | 1471 | 1,589 | 1,697 | 1,801 | 1,898 | .. | .. | .. | .. |
| 4·75 | .. | .. | 1156 | 1250 | 1337 | 1492 | 1636 | 1,767 | 1,888 | 2,004 | 2,113 | .. | .. | .. | .. |
| 5·00 | .. | .. | 1282 | 1385 | 1480 | 1654 | 1812 | 1,956 | 2,088 | 2,221 | 2,340 | 2,455 | .. | .. | .. |
| 5·25 | .. | .. | .. | 1527 | 1633 | 1824 | 2000 | 2,159 | 2,307 | 2,448 | 2,580 | 2,706 | .. | .. | .. |
| 5·50 | .. | .. | .. | 1676 | 1792 | 2000 | 2192 | 2,364 | 2,528 | 2,733 | 2,880 | 3,026 | .. | .. | .. |
| 5·75 | .. | .. | .. | 1829 | 1956 | 2185 | 2395 | 2,586 | 2,762 | 2,932 | 3,085 | 3,240 | .. | .. | .. |
| 6·0 | .. | .. | .. | .. | 2140 | 2380 | 2609 | 2,820 | 3,012 | 3,196 | 3,366 | 3,531 | 3,687 | 3,842 | .. |
| 6·5 | .. | .. | .. | .. | 2508 | 2800 | 3068 | 3,316 | 3,540 | 3,760 | 3,955 | 4,154 | 4,336 | 4,514 | .. |
| 7·0 | .. | .. | .. | .. | 2912 | 3252 | 3564 | 3,852 | 4,112 | 4,364 | 4,580 | 4,804 | 5,015 | 5,221 | .. |
| 7·5 | .. | .. | .. | .. | .. | 3728 | 4088 | 4,412 | 4,720 | 5,004 | 5,264 | 5,518 | 5,760 | 5,996 | 5,438 |
| 8·0 | .. | .. | .. | .. | .. | 4228 | 4644 | 5,016 | 5,448 | 5,688 | 5,980 | 6,283 | 6,559 | 6,818 | 6,223 |
| 8·5 | .. | .. | .. | .. | .. | 4780 | 5240 | 5,660 | 6,040 | 6,416 | 6,768 | 7,100 | 7,413 | 7,756 | 7,074 |
| 9·0 | .. | .. | .. | .. | .. | .. | 5884 | 6,356 | 6,788 | 7,204 | 7,592 | 7,961 | 8,311 | 8,653 | 8,009 |
| 9·5 | .. | .. | .. | .. | .. | .. | 6544 | 7,068 | 7,552 | 8,016 | 8,452 | 8,864 | 9,253 | 9,633 | 8,979 |
| 10·0 | .. | .. | .. | .. | .. | .. | 7248 | 7,824 | 8,352 | 8,885 | 9,360 | 9,820 | 10,249 | 10,670 | 9,996 |
| 11·0 | .. | .. | .. | .. | .. | .. | .. | 9,466 | 10,111 | 19,733 | 11,314 | 11,844 | 12,384 | 12,863 | 11,073 |
| 12·0 | .. | .. | .. | .. | .. | .. | .. | 11,280 | 12,050 | 12,784 | 13,464 | 14,124 | 14,748 | 15,368 | 13,379 |

Ordinarily, in the merchant service, the area of chimney is about a fifth to a quarter of the grate area, whilst in the Navy they are seldom larger that 0·14, and vary from that to 0·11, according to the air-pressure intended to be used.

The thickness of funnel plates may be approximately as follows:—

| | | |
|---|---|---|
| Upper plates | ... | ·1-inch + (·01 inch for each foot of diameter). |
| Middle ,, | ... | ·125 ,, + (·01 ,, ,, ). |
| Lower ,, | ... | ·15 ,, + (·01 ,, ,, ). |

Funnels should be suitably stiffened by angle, tee, or channel rings or hoops, and large ones cross-stayed and stiffened longitudinally.

The funnels of Naval ships are made considerably lighter, the upper plates being usually $6\frac{3}{4}$ lbs. per sq. foot, even for the largest sizes.

## Table LXXXVIIIa.—Pitch, &c., of Riveting for Funnels, Casings, &c. (Admiralty work).

| Description. | Thickness of Plate. | Diameter of Rivet. | Pitch of Rivet. |
|---|---|---|---|
| | Inch. | Inch. | Inches. |
| Uptakes, . . . | $\frac{3}{16}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{4}$ | $\frac{1}{2}$ | 2 |
| | $\frac{5}{16}$ | $\frac{5}{8}$ | $2\frac{1}{2}$ |
| | $\frac{3}{8}$ | $\frac{5}{8}$ | $2\frac{1}{2}$ |
| Funnels, . . . . | $\frac{1}{4}$ | $\frac{1}{2}$ | 3 |
| | $\frac{3}{16}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{8}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{16}$ | $\frac{1}{4}$ | 2 |
| Funnel casing, . . | $\frac{3}{16}$ | $\frac{3}{8}$ | 3 |
| | $\frac{1}{8}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{16}$ | $\frac{1}{4}$ | 2 |
| Deck casing, . . . | $\frac{3}{16}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{8}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{16}$ | $\frac{1}{4}$ | 2 |
| Cowls and trunks (exposed to weather), . . . | $\frac{1}{8}$ | $\frac{3}{8}$ | 2 |
| | $\frac{1}{16}$ | $\frac{1}{4}$ | 2 |
| Screens and ventilating trunks, . . . | $\frac{1}{8}$ | $\frac{3}{8}$ | 4 |
| | $\frac{1}{16}$ | $\frac{1}{4}$ | 3 |
| Other work, . . . | $\frac{3}{16}$ | $\frac{3}{8}$ | 2 to 3 |
| | $\frac{1}{8}$ | $\frac{3}{8}$ | 2 to 3 |
| | $\frac{1}{16}$ | $\frac{1}{4}$ | 2 |

When a number of boilers are served by the same funnel, and some of the furnaces are much further away from the base of the funnel than others (as is sometimes unavoidably the case in Naval vessels) dampers for regulating the force of the draught should be fitted in each "leg" of the uptakes.

Similarly,—proper arrangements (flues, screens, &c.) should be fitted to distribute the air in the stoke-holes, when the "closed stoke-hole" system of forced draught is used.

Smoke-box doors should be made "three thick,"—and arrangements provided to obtain a good current of air between the inner and central plates, whilst the space between the central and outer plates should be filled with some good non-conducting substance.

**Forced draught.**—In the Navy the "closed stoke-hole" system of forced draught is now generally used; as the name implies, the whole stoke-hole is made air-tight, and the fans force air down into it,—the men working under pressure and entering and leaving through air locks.

Of the "closed ash-pit" systems, the Howden is the one now generally adopted in the mercantile marine. In this system air is led by pipes from the fans to each furnace separately, and can thus be shut off from any one before opening the door, so that the rush of cold air, which is such an objectionable feature in the closed stoke-hole system, is entirely avoided. The Howden also heats the air supply to 180° to 200° by passing it amongst a number of tubes placed in the uptake, and thus no doubt materially increases the efficiency of the furnace, since this increase of temperature so facilitates combustion as to give a gain much in excess of the actual amount of heat stored in the air.

Messrs Howden state that the horse-power per square foot of grate obtained from boilers fitted on this system varies from about 15 on tramp steamers working with very poor bunker coal, to over 26 on cruisers using best Welsh steam coal; and that the corresponding heating surfaces per I.H.P. would be about 2·8 square feet and 1·6 square feet.

As regards coal consumption, Messrs Howden consider that—in the case of triple engines working with 160 lbs. to 180 lbs. steam pressure —1·2 lbs. of Cardiff or equally good coal per I.H.P. per hour should not be exceeded in regular work at sea.

The air pressure used in the ash-pits varies from ½ inch to 1½ inch, according to the circumstances of the case.

An air temperature of about 300° is now aimed at, and the effect of the abstraction of this heat from the waste gases is to materially lower their temperature. The air tubes are, as a rule, 2¾ inches external diameter and 14 L.S.G. in thickness. Their length varies from 30 inches to 60 inches.

Coal consumptions up to 50 lbs. per square foot of grate per hour have been effected without the slightest trouble.

Mr Howden claims for his system the greatly reduced space and weight which must be allotted to boilers, the reduced coal consumption, and the very uniform temperature of furnaces, all of which latter tend to reduce repairs and to lengthen the life of the boiler.

The great advantage of artificial or forced draught is that at all times it is completely under control, and quite independent of wind or weather, thus ensuring a uniform efficiency of furnace with all ordinary qualities of coal, and any rate of combustion that may be required. In Naval ships this is a necessity,—since the chimney may be entirely destroyed at any moment.

Pressures of air in stokeholds, &c., are usually reckoned in inches of water (often called for shortness "inches" simply), and are measured by a "water-gauge," **U** form, partly filled with water, and having the open end of one leg exposed to the pressure to be measured, and the other to the outer air. The pressure is indicated by the difference in level of the water in the two legs.

**Stokehold ventilators.**—The downcast pipes with natural draught should have an aggregate transverse area of 0·45 square inch per lb. of fuel burnt per hour; or 0·675 square inch per I.H.P. in cargo boats, 0·75 in express steamers; 0·62 is sufficient for turbine-driven ships.

The area of the cowl mouth should be not less than

| | | | | | |
|---|---|---|---|---|---|
| 1·35 | square inch | per lb. fuel | per hour in | 10 | knot ships. |
| 1·24 | ,, | ,, | ,, | 12½ | ,, |
| 1·13 | ,, | ,, | ,, | 15 | ,, |
| 0·93 | ,, | ,, | ,, | 20 | ,, |
| 0·78 | ,, | ,, | ,, | 25 | ,, |

Table LXXXIX. shows the results of tests of the various modern boilers with oil and coal fuel.

Table XC. shows the rates of combustion and of evaporation usual on sea trials of various classes of vessel under the conditions named.

With inferior coal, supplied at foreign stations, the figures in columns 5 and 6 may be reduced as much as 20 per cent., whilst with picked coal and highly skilled stokers, they may be increased nearly 10 per cent.

The feed temperatures usual in the respective classes of vessels are assumed, and allowance should be made for any special feed-heating.

The evaporative efficiency of the boilers of cruisers and battleships appears lower than it really is, as a large amount of steam is consumed by the numerous auxiliary engines and for domestic purposes, but is left out of consideration here; for quantities of steam so consumed see Tables XCII. and XCIII.

By way of a comparison with Table XC., it may be noted that a goods locomotive consumes about 40 lbs. of coal per foot of fire-grate per hour, and an express passenger engine from 65 to 80 lbs.; and, as the evaporative efficiency of the locomotive boiler is high, the water evaporated per square foot of grate per hour may be taken as about 420 lbs. for the goods engine, and from 650 to 750 lbs. for the passenger engine.

## Table LXXXIX.—Results of Trials of

| | U.S.A. Hohenstein W.T. Boiler. | | | B. | C. 12·5 diam., &c. | | | Yarrow Naval, Light Weight. |
|---|---|---|---|---|---|---|---|---|
| | Natural Draught. | Fcd. Dght. =2·1 ins. | Fcd. Dght. =1·0 in. | 12 ft. diam., 11 ft. long. 2 Furnaces. | Nat. Dght. | Nat. Dght. | Fcd. Dght. =1·5 ins. | |
| Type of boiler . . | Water tube | Water tube | Water tube | Cylindrical | Cylindrical | Cylindrical | Cylindrical | Water tube |
| Total heating surface, sq. ft. | 2,130 | 2,130 | 2,130 | 1,200 | 1,695 | 1,695 | 1,695 | 8,300 |
| Grate area, sq. ft. . | 50·14 | 50·14 | 50·14 | .. | 40 | 40 | 40 | .. |
| Ratio of T.H.S. to grate area. | 42·5 | 42·5 | 42·5 | .. | 42·3 | 42·3 | 42·3 | .. |
| Working pressure, lbs. | 275 | 275 | 275 | 140 | 120 | 120 | 120 | 250 |
| Fuel, kind of . . | Pocahontas coal | Coal | Texas oil | Texas oil | New-castle coal | Texas oil | Texas oil | Texas oil |
| Fuel value in B.T.U. | 14,992 | 14,992 | 19,481 | 19,000 | 14,500 | 19,000 | 19,000 | 19,000 |
| Engines supplied . | .. | .. | .. | .. | .. | .. | .. | Turbines |
| Engines, horse-power, per boiler. | .. | .. | .. | .. | .. | .. | .. | 8,900 |
| Fuel consumed per hr., lbs. | 1,037 | 2,674 | 1,969 | 820 | 974 | 633 | 1,222 | 9,300 |
| Fuel consumed per sq. ft. of grate. | 20·68 | 50·33 | .. | .. | 24·35 | .. | .. | .. |
| Fuel consumed per sq. ft. of T.H.S. | 0·49 | 1·26 | 0·92 | 0·68 | 0·58 | 0·37 | 0·72 | 1·12 |
| Fuel consumed per H.P. | .. | .. | .. | .. | .. | .. | .. | 1·033 |
| Water evaporated per hour, lbs. | 8,769 | 21,549 | 21,309 | 10,917 | 7,558 | 7,756 | 14,591 | 107,000 |
| Water evaporated per lb. of fuel. | 8·46 | 8·06 | 10·82 | 13·31 | 7·76 | 12·25 | .. | 11·70 |
| Water evaporated per lb. of fuel from and at 212°. | 9·645 | 9·38 | 12·37 | 16·13 | 9·31 | 14·45 | 14·06 | 13·68 |
| Water evaporated per sq. ft. of T.H.S. | 4·12 | 10·12 | 10·00 | 9·09 | 4·46 | 4·58 | 8·61 | 12·84 |
| Water evaporated per H.P. | .. | .. | .. | .. | .. | .. | .. | 12·05 |
| Total heating surface per H.P. | .. | .. | .. | .. | .. | .. | .. | 0·93 |
| Efficiency per cent. . | 62·11 | 60·41 | 61·34 | 82·08 | 62·00 | 73·46 | 71·48 | 71·47 |
| Total weight of boiler, &c., per ft. T.H.S., lbs. | 25·4 | 25·4 | .. | .. | .. | .. | .. | 13·34 |
| Total weight of boiler, &c., per H.P., lbs. | .. | .. | .. | .. | .. | .. | .. | 12·45 |
| Total weight of boiler, &c., lbs. water evaporated per hour. | 6·16 | 2·51 | 2·51 | .. | .. | .. | .. | 1·035 |

## Boilers at Full Power.

| Thornycroft, Light Naval, Fcd. Dght. | White-Forster Naval Boilers, Light Naval. | | | Babcock-Wilcox, U.S.A. Naval Boiler. | | Miyabara, Japanese Naval Boiler. | | Ordinary Mercantile. 16·2 ft. diam., 11·5 ft. long. | Niclausse Naval, Fcd. Dght. | Belleville Naval, Fcd. Dght. |
|---|---|---|---|---|---|---|---|---|---|---|
| | Fcd. Dght. | Nat. Dght. | Fcd. Dght. | Nat. Dght. | Fcd. Dght. | Fcd. Dght. | Fcd. Dght. | | | |
| Water tube | Water tube | Water tube | | Water tube | | Water tube | | Cylindrical | Water tube | Water tube |
| 7,930 | 7,500 | 4,000 | | 5,353 | | 1,912 | | 2,994 | 8,000 | 4,099 |
| .. | 125·0 | .. | | 119 | | 53·91 | | 68·6 | 231·8 | 146 |
| .. | 60·00 | .. | | 44·98 | | 35·47 | | 43·67 | 34·5 | 28·1 |
| 265 | 220 | 203 | 208 | 203 | 210 | 234 | | 180 | 144 | 287 |
| Texas oil | Coal | Texas oil | | Pocahontas coal | | Welsh coal | Californian oil | Coal | Coal | Coal |
| 19,000 | 13,800 | 19,000 | | 15,000 | | 14,800 | 18,500 | 14,200 | 14,000 | 14,800 |
| Turbines | Turbines | .. | | Turbines | | .. | .. | Recpro. turbines | Triple comp. | Triple comp. |
| 8,600 | 5,000 | .. | | 612 | 3,484 | .. | .. | 1,550 | 2,773 | 1,696 |
| 8,810 | 9,250 | 1,145 | 3,800 | 1,365 | 5,549 | 2,021 | 1,590 | 1,920 | 5,934 | 3,138 |
| .. | 74·0 | .. | .. | 11·48 | 46·73 | 37·41 | 29·49 | 26·24 | 25·60 | 21·49 |
| 1·110 | 1·23 | 0·287 | 0·950 | 0·255 | 1·037 | 1·057 | 0·831 | 0·641 | 0·742 | 0·765 |
| 1·025 | 1·85 | .. | .. | 2·23 | 1·60 | .. | .. | 1·24 | 2·14 | 1·85 |
| 105,500 | 67,500 | 15,750 | 45,410 | 14,517 | 46,686 | 15,450 | 14,983 | 18,786 | 50,607 | 28,460 |
| 11·92 | 7·30 | 13·75 | 11·94 | 10·66 | 8·87 | 7·51 | 10·73 | 9·79 | 8·53 | 9·09 |
| 14·00 | 8·58 | 16·70 | 14·50 | 12·47 | 10·37 | 9·16 | 13·09 | 11·76 | 10·34 | 10·63 |
| 13·24 | 9·00 | 3·93 | 11·35 | 2·76 | 8·72 | 8·08 | 7·84 | 6·27 | 7·67 | 6·94 |
| 12·27 | 13·50 | .. | .. | 23·77 | 13·42 | .. | .. | 12·12 | 18·25 | 16·82 |
| 0·92 | 1·50 | .. | .. | 8·74 | 1·54 | ... | .. | 1·93 | 1·35 | 2·41 |
| 71·7 | 60·6 | 84·9 | 73·7 | 80·3 | 66·8 | 59·5 | 68·3 | 80·0 | 71·3 | 70·8 |
| 13·84 | 12·83 | .. | .. | 24·27 | 24·27 | .. | .. | .. | .. | 41·35 |
| 12·76 | 19·26 | .. | .. | 212 | 37·4 | .. | .. | .. | .. | 103·4 |
| 1·05 | 1·43 | .. | .. | 8·79 | 2·78 | .. | .. | .. | .. | 6·16 |

## Table XC.—Rates of Combustion and Evaporation. (Tank Boilers.)

| Description of vessel. | Type of boiler, grate, &c. | Nature and force of draught. | Pounds of coal per ft. of grate per hour. | Pounds of water per pound of coal. | Pounds of water per ft. of grate per hour. |
|---|---|---|---|---|---|
| Ordinary Merchant Steamer | Ordinary cylindrical, long bars | Chimney only | 15 | 9·15 | 137 |
| ,, ,, | Ordinary cylindrical, short bars | ,, ,, | 20 | 9·25 | 185 |
| Passenger or Mail Steamer | Ordinary cylindrical medium bars | Chimney, and ·5–·75 in. water | 25–30 | 9–8·75 | 225–262 |
| Cruiser or Battleship | Ordinary cylindrical, very long bars | Chimney, and ·5 in. water | 30 | 8 | 240 |
| ,, ,, | ,, ,, | Chimney, and 2 in. water | 42 | 7·75 | 325 |
| Torpedo Gunboat | Locomotive type, square grate | Chimney, and 2·5 in. water | 55 | 7·2 | 396 |
| Torpedo-Boat | ,, ,, | 3 in. water | 67 | 6·8 | 456 |
| ,, | ,, ,, | 6 in. water | 96 | 6·6 | 633 |
| Paddle Steamer | Rectangular box, 7 foot bars | Chimney only | 22 | 7·7 | 170 |
| ,, | Rectangular box, 6 foot 6 inch bars, brass tubes | Chimney, and ·5 in. water | 27 | 7·2 | 195 |
| ,, | Ordinary cylindrical, 6 foot 6 inch bars, brass tubes | Chimney, and ·5 in. water | 25 | 8·8 | 220 |

## Table XCI.—Express Boilers. Particulars of Surface, Weight, &c.

| Name of Ship. | Type of Boiler. | Inventor. | Number of Boilers. | Total Grate Area. | Total Heating Surface. | Ratio Heating Surface to Grate. | Working Pressure. | Total Weight of Boilers, etc. | Total Weight per sq. ft. of Grate. | Total Weight per I.H.P. | Total I.H.P. on Trial. | Air Pressure in Stokehold. | I.H.P. per sq. ft. of Grate. | I.H.P. per ton of Boilers. | Coal per I.H.P. per hour. | Coal per sq. ft. of Grate per hour. | Heating Surface per I.H.P. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Sq.Ft. | Sq.Ft. | | Lbs. | Tons. | | Lbs. | | Ins. | | | Lbs. | Lbs. | Sq.Ft. |
| H.M.S. :— | | | | | | | | | | | | | | | | | |
| Rke. . | Loco. | Laird | 2 | 126 | 4,639 | 36·8 | 140 | 85·2 | 1515 | 70·0 | 2,740 | 2·30 | 21·7 | 32·1 | .. | .. | 1·69 |
| Kta. . | ,, | Belliss | 2 | 190 | 5,330 | 28·0 | 150 | 95·6 | 1127 | 55·7 | 3,840 | 1·70 | 20·2 | 40·4 | .. | .. | 1·3 |
| Gmr. . | ,, | Sheerness | 2 | 183 | 5,654 | 30·8 | 150 | 103·4 | 1267 | 63·4 | 3,654 | 2·70 | 19·9 | 35·4 | .. | .. | 1·5 |
| Spy. . . | W. T. | Thornycroft | 8 | 204 | 17,700 | 86·3 | .. | 107·2 | 1177 | 51·0 | 4,703 | 1·70 | 23 | 43·9 | .. | .. | 3·7 |
| Hvk. . | Loco. | Yarrow | 2 | 100 | 4,656 | 46·5 | 180 | 58·0 | 1299 | 37.0 | 3,497 | 3·66 | 34·9 | 60·0 | 2·59 | 90·6 | 1·33 |
| Hnt. . | W. T. | Yarrow | 8 | 154·4 | 8,216 | 53·2 | 180 | 49·0 | 711 | 28·0 | 3,884 | 1·60 | 25·1 | 79·1 | 2·4 | 60·0 | 2·11 |
| Dng. . | ,, | Thornycroft | 3 | 189 | 8,892 | 41·7 | 210 | 56·0 | 644 | 30·0 | 4,208 | 3·37 | 22·2 | 75·0 | 3·49 | 77·8 | 2·11 |
| Frt. . . | ,, | Laird<br>Normand | 4 | 154 | 8,812 | 52·7 | 175 | 64·2 | 900 | 32·0 | 4,484 | 4·10 | 29·9 | 69·7 | 3·88 | 113 | 1·96 |
| Jns. . . | ,, | Reed | 4 | 209 | 10,164 | 48·6 | 210 | 70·6 | 757 | 41·0 | 3,885 | 2·39 | 18·5 | 54·9 | 3·15 | 58·6 | 2·61 |
| Hgty. . | ,, | Yarrow | 8 | 179 | 8,203 | 45·8 | 200 | 62·1 | 777 | 33·0 | 4,208 | 2·30 | 23·5 | 67·7 | 2·72 | 63·9 | 1·94 |
| Cnflt. . | ,, | White | 3 | 212·5 | 11,250 | 52·9 | 185 | 82·0 | 875 | 38·0 | 4,931 | 4·20 | 23·2 | 59·4 | .. | .. | 2·32 |
| Strgn. . | ,, | Blechynden | 4 | 176 | 10,022 | 57·0 | 200 | 62·2 | 791 | 32·0 | 4,367 | 3·1 | 24·8 | 70·1 | 3·29 | 81·6 | 2·29 |
| Ql. . . | ,, | Normand | 4 | 210 | 11,945 | 56·8 | 220 | 75·0 | 800 | 26·0 | 6,347 | 4·25 | 30·2 | 84·6 | 2·48 | 74·9 | 1·88 |
| Dspt. . | ,, | Thornycroft | 3 | 196 | 11,855 | 60·5 | 220 | 66·0 | 754 | 25·0 | 5,795 | 3·45 | 29·5 | 87·8 | 2·37 | 70·1 | 2·04 |
| Str. . . | ,, | Reed | 4 | 255 | 13,384 | 52·5 | 250 | 89·4 | 785 | 32·0 | 6,318 | 3·32 | 24·7 | 70·6 | 2·53 | 62·7 | 2·11 |
| Gpy. . | ,, | Yarrow | 4 | 245 | 13,561 | 55·3 | 250 | 68 | 622 | 23·0 | 6,523 | 4·4 | 26·6 | 95·9 | 2·51 | 66·8 | 2·07 |
| An. . | ,, | Thornycroft | 4 | 236 | 13,400 | 56·8 | 230 | 71 | 674 | 26·0 | 6,153 | 3·6 | 26·0 | 86·6 | 2·59 | 67·5 | 2·17 |
| Al. . . | ,, | White-Forster | 6 | 750 | 45,000 | 60·0 | 220 | 268 | 801 | 20·0 | 30,000 | .. | 40·0 | 112 | 1·86 | 74 | 1·50 |

**Table XCII.—H.M.S. "Diana." Water Consumption Trials, showing Cost of Auxiliaries. Cylindrical S.E. Boilers. 155 lbs. Working Pressure. Made in May and June, 1899, for Admiralty Boiler Committee.**

| Description of Trial. | Percentage of Full Power. | Steam Pressure. | | | Lbs. of Water per I.H.P. per Hour. | | | Lbs. of Coal per I.H.P. per Hour. | Remarks. |
|---|---|---|---|---|---|---|---|---|---|
| | | Boilers. | Engines, Starboard. | Engines, Port. | Main Engines. | Auxiliary. | All Purposes. | | |
| 30 hours at 800 I.H.P. | 8·3 | 101 | 96 | 97 | 20·59 | 5·68 | 26·27 | 2·82 | |
| 12 ,, 1600 ,, | 16·6 | 124 | 120 | 121 | 18·94 | 4·21 | 23·15 | 2·52 | Cut-off in H.P. cylinder 38 per cent. |
| 30 ,, 1600 ,, | 16·6 | 117 | 113 | 113 | 18·26 | 3·29 | 21·55 | 2·52 | Cut-off in H.P. cylinder 44·5 per cent. |
| 30 ,, 4800 ,, | 50·0 | 133 | 128 | 128 | 17·09 | 2·14 | 19·23 | 2·02 | |
| 30 ,, 6400 ,, | 66·6 | 138 | 129 | 129 | 17·9 | 2·56 | 20·46 | 2·38 | |
| 8 ,, 8000 ,, | 83·3 | 150 | 144 | 144 | 17·75 | 1·94 | 19·69 | 2·39 | |

**Table XCIIa.—R.M.S. "Lusitania." Water Consumption Trials, showing the demands for Auxiliary Machinery and Domestic Purposes.**

| Speed in Knots. | 15·75. | 18·00. | 21·00. | 23·00. | 25·40. |
|---|---|---|---|---|---|
| Shaft horse-power . . . . . . | 13,400 | 20,500 | 33,000 | 48,000 | 68,850 |
| Consumption per hour of steam of main engines . . . . . . . lbs. | 284,500 | 353,600 | 493,300 | 668,300 | 879,500 |
| Consumption per hour of steam of auxiliaries, &c. . . . . lbs. | 93,500 | 100,900 | 112,700 | 127,500 | 149,700 |
| Consumption per hour of steam, total lbs. | 338,000 | 454,500 | 606,000 | 795,800 | 1,029,200 |
| Consumption per hour of steam of main engines per S.H.P. hour . . . | 21·23 | 17·24 | 14·91 | 13·92 | 12·77 |
| Consumption per hour of steam of auxiliaries, &c., per S.H.P. hour . | 6·97 | 4·92 | 3·41 | 2·65 | 2·17 |
| Consumption per hour of steam, total per S.H.P. hour . . . . . . | 28·20 | 22·16 | 18·32 | 16·57 | 14·94 |
| Feed water temperature, F.° . . . . | 200 | 200 | 200 | 200 | 200 |
| Consumption of coal per S.H.P. hour . | 2·76 | 2·17 | 1·80 | 1·62 | 1·46 |

Table XCIII.—Consumption of Water (Steam) and Fuel on Trials of Various Ships.

| Ship. | Engines. | | Boilers. | | Per I.H.P. hour. Water for all purposes. | | | Per I.H.P. hour. Water Main Engines only. | | | Per I.H.P. hour. Fuel. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Sort of. | I.H.P. | Sort of. | Press. | Full Power. | $\frac{4}{5}$ to $\frac{2}{3}$ Power. | Low Power. | Full Power. | $\frac{4}{5}$ to $\frac{2}{3}$ Power. | Low Power. | Full Power. | $\frac{4}{5}$ to $\frac{2}{3}$ Power. | Low Power. |
| Cargo s.s. | Triple | 1,000 | S.E. Cylindrical | .. | 17·50 | 18·52 | 18·81 | .. | .. | .. | .. | .. | .. |
| " | G. Turb. | 1,000 | " " | .. | 14·75 | 16·06 | 17·28 | .. | .. | .. | .. | .. | .. |
| Express s. | Turbines | 65,000 | D.E. Cylindrical | 195 | 14·94 | 22·16 | 26·53 | 12·77 | 17·24 | 21·23 | 1·43 | 2·01 | 2·52 |
| Passenger s. | Mixed | 7,000 | " " | 195 | 13·66 | 13·79 | .. | 11·95 | 12·80 | 13·60 | .. | .. | .. |
| Channel express | Turbine | 14,700 | " " | .. | .. | .. | .. | .. | .. | .. | 1·87 | .. | .. |
| " " | Triples | 7,000 | D.E. & S.E. Cylr. | 150 | .. | .. | .. | .. | .. | .. | 2·26 | 1·95 | .. |
| Cruiser B. | Triple | 15,500 | .. | .. | .. | .. | .. | .. | .. | .. | 1·92 | 2·47 | 2·89 |
| " S. | Turbines | 19,000 | .. | .. | .. | .. | .. | .. | .. | .. | 1·81 | 1·78 | 2·68 |
| " P. | " | 11,500 | Water tube | 246 | 14·9 | 15·50 | 21·8 | .. | .. | .. | 1·82 | .. | .. |
| " Ch. | Triple | 23,500 | Cylindrical | 190 | 19·08 | 18·28 | 21·36 | 16·4 | .. | .. | 1·99 | 2·04 | 2·00 |
| " Jp. | Turbines | .. | .. | .. | 15·05 | 16·50 | 21·07 | 13·77 | 15·6 | 20·57 | .. | .. | .. |
| " G. | " | 25,000 | Cylr. & water tube | .. | .. | .. | .. | .. | .. | .. | 1·59 | 1·65 | .. |
| Battleship A. | Triples | 18,600 | " " | .. | 19·9 | 16·95 | .. | .. | 15·31 | .. | 2·03 | 1·85 | 1·88 |
| Cruiser Aa. | " | 16,800 | Water tube | .. | .. | .. | .. | .. | .. | .. | 1·76 | 17·4 | 1·98 |
| " Hs. | " | 10,300 | Belvl. water tube | 228 | .. | .. | .. | .. | .. | .. | 1·467 | 1·54 | .. |
| " A. | Turbines | 14,000 | Cylr. & water tube | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| Channel express | " | 6,700 | D.E. Cylindrical | 160 | 17·1 | .. | .. | .. | .. | .. | 1·71 | .. | .. |
| " " | G. Turb. | 5,000 | " " | 160 | 14·3 | .. | .. | .. | .. | .. | .. | .. | .. |
| Cargo s.s. | Triples | 650 | S.E. Cylindrical | 180 | 13·35 | .. | .. | 13·35 | .. | .. | 1·46 | .. | .. |
| Passenger 4-s.s. | Mixed | 10,800 | " " | 180 | 15·17 | 12·37 | .. | .. | 10·65 | .. | .. | .. | .. |
| Battleship | Turbines | 40,500 | Water tube | 210 | 13·4 | 15·23 | 23·77 | 12·18 | 13·09 | 16·82 | 1·59 | 1·72 | 2·23 |

# MARINE BOILERS—KINDS OF.

**Boilers for ship purposes** are divided into two classes known as tank and tube.

**The tank boiler** now used is cylindrical in form and contains within it the water and the apparatus for heating and evaporating it.

**The tube boiler** consists almost wholly of tubes, and generally there is a cylindrical vessel at the top, called the receiver, into which the tubes deliver the steam generated. If the tube delivers below the normal water level of the receiver they are said to be *drowned*, as distinct from those which deliver their contents above the water level, as was usual with the Thornycroft boilers. Tube boilers are also divided into two sub-divisions, each distinguished by the size of the tubes. Those made up to 1½-diameter are called the small tube variety, while those with tubes of greater diameter are called the large tube variety. The water is within the tubes, and the fire-place is an independent apparatus below the boiler; the heat is applied to the outer surface of the tubes.

**The tank boiler** contains much more water than the tube boiler, and is the one almost exclusively in use in all kinds of mercantile ships. In some few such ships, where very high speed and consequent very light weight is desirable, water tube boilers are used.

**The water tube boiler** is used very largely in all warships, as being lighter and safer against shot and shell. It is not, however, exclusively so, as at one time was the case. As a result of the investigations of the British Admiralty Boiler Committee, warships are now * supplied with about one-third of the power of cylindrical boilers, as they are better suited for and more economic when cruising at low speeds. The water tube boiler can be brought into action much quicker and stand rougher usage in the process than the tank, for steam can be raised in twenty minutes to half an hour after lighting fires on many water tube boilers, whereas several hours are necessary for raising steam in the tanks; in fact, it is not uncommon to allow six or more hours. There is very little difference in cost of a tube boiler for 300 lbs. working pressure over that of one for 150; a tank boiler of large size cannot be made for the former pressure, and for 230 lbs. is a much heavier and costlier one than if made for 150 lbs. On the other hand, the life of the tank boiler is *pari passu* longer than that of the water tube; and much longer than that of the express small tube boilers.

**Cylindrical boilers** are usually "horizontal," but there are one or two vertical forms used for other than auxiliary purposes with satisfaction. The Cochran boiler, and others similar to it, are useful and convenient for many purposes, especially where floor space is limited.

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* Latterly there has been a reversion to the water tube boiler exclusively.

The Scotch haystack boiler was a very useful one for river steamers of high speed when moderate pressures were the rule.

**The horizontal cylindrical boiler** in general use has two, three, or four furnaces in the single-ended variety, and double those numbers in the double-ended ones of same diameters. They are now made up to 19 feet diameter, and the double-ended as much as 22 feet long. The working pressure is sometimes as high as 235 lbs., but generally 210 to 220 for quadruple expansion engines, and 180 to 205 for triples. In the Navy the pressure for cylindrical boilers is usually 190 lbs.

For such large diameters and high pressures, high tensile steel is generally used; 36 tons is the common test per square inch, while special steel having an ultimate strength of 40 tons can be used with advantage. There is no shock on such steel, no reversal of stresses, and the application and removal of load is most gentle.

The longitudinal seams are generally of the special treble-riveted type, but may be, as they are largely on the Continent and in America, special quadruple with vandyked edges to the butt straps, whereby as much as 94 per cent. of solid plate is obtained. The end circumferential seams are always double riveted for large sizes and high pressures, the middle seams of such double-ended ones are treble riveted.

**The number of combustion chambers** varies in practice, at one time even quite large boilers had sometimes only one chamber common to all six furnaces; sometimes a chamber common to all the furnaces of one end of a double-ended boiler. Sometimes, and this was quite general practice, there was a combustion chamber common to each pair of opposite furnaces, so that a six-furnace boiler would have three chambers. Of late years the tendency has been to have a separate chamber for each furnace: this is expensive and makes a heavy boiler, but is a safe one and one more easily worked well—especially at cleaning fire time than the others.

**The gunboat boiler**, in which the furnaces are at one end, the tubes at the other, and the combustion chamber between them, was largely used in the Navy; of late years it has been adopted with advantage in shallow passenger steamers where deck space is important and hold space free. It is a good steaming boiler and for small units compares very favourably with the ordinary cylindrical, but requires practically two stoke-holds, the second at the back to permit of cleaning or removing the tubes.

**The locomotive boiler** was formerly extensively used in small craft of all kinds, and in the Navy for very high powers. The water tube boiler has, however, quite superseded it.

**The double-ended boiler** is naturally the cheaper and lighter, as may be seen by referring to Table XCVIII. It is the one generally used for high powers in express steamers; but it requires more careful handling than similarly sized single-ended ones, especially when raising steam, as the bottom will remain quite cool after steam has formed on the top, unless means are adopted to ensure circulation.

**Water tube boilers** now in general use are much more limited in type and design than formerly. Time has tried them all, and comparatively few have continued to retain the confidence of marine engineers.

**Of the large tube boilers** there remains the Babcock & Wilcox, so much used in the British and American Navies; the Niclausse in the French Navy, and the Dürr in the German. The Belleville has practically gone, owing to having got a bad name—perhaps worse than it deserved,—but largely to its great cost and expense of upkeep. They all have straight tubes, are simple in design and construction, and easy to work. Their evaporative efficiency is good with good stoking, and they can be forced to a considerable amount without injury—especially the Niclausse, which has a positive circulation through each and every tube.

**The Yarrow type of boiler** is made now generally for the big ships with comparatively large tubes, so as to entitle them almost to be called large tube boilers. With their straight or nearly straight tubes approaching the vertical, and the simplicity of their construction, they can stand a lot of forcing and rough usage, and are quite efficient evaporators.

**The Hohenstein boiler** also is one liked in the U.S. America; while in Japan the "Miyabara" boiler, somewhat like the Hohenstein in general design, is a favourite; both have straight tubes, and are simple in construction and easily worked.

**Of small tube boilers** there are, in addition to the Yarrow, the Normand and others of the same type, differing in detail. The double-ended variety of M. Sigaudy; the Mumford modification of it, the Blechynden variation from the Yarrow, and the White-Forster variation of the Blechynden are good boilers. The Reed, with curved tubes drowned, the Thornycroft "Daring," with undrowned tubes, the Mumford, with curved tubes, are good, especially for small craft; as is also the boiler of Simpson of Dartmouth. The Admiralty continue to show a strong predilection for straight or nearly straight tubes for all kinds of water tube boilers; on the other hand, there is more scope for design, greater economy of space and flexibility of structure with the curved tubes, and with the moderate amount, as in the Normand type, little fault is to be found. The freer the flow in the tubes, the better both for efficiency and wear. With a strong positive flow there is no priming, and no danger from lodgment of bubbles nor any deposit of solids on the heating surface.

No doubt the troubles with the Belleville, and the controversies following them, largely account for the prejudice against all water tube boilers in the mercantile marine; there is, besides this, the fact that while these boilers are not necessarily wasteful of fuel, and are, when carefully worked, as economic as ordinary tank boilers, they can be made, in careless hands, more extravagant consumers than tank boilers can be under similar circumstances. On land, for central electricity stations, the water tube boiler is the favourite as responding to sudden and unexpected calls, and is found to be, with mechanical stoking, quite efficient and economical.

**The total heating surface** of a cylindrical marine boiler may be estimated with quite a fair degree of approximation by

**Rule 245.** **Total heating surface** $= D^2 \times L \times K$ square feet.

D is the diameter and L is the length of shell in feet. When the tubes are of moderate diameter ($2\frac{3}{4}$ to $3\frac{1}{2}$ inches) and spaced as usual in the mercantile marine, and with good room for examination and cleaning, $K = 0{\cdot}87$ for single-ended and $1{\cdot}08$ for double-ended boilers. When the tubes are smaller ($2\frac{1}{2}$ to 3), and as many of them as can be got in consistent with satisfactory service, K may then be 1·0 for single and 1·2 for double-ended boilers.

Another method of obtaining roughly the total heating surface of a boiler is as follows:—

**Rule 245a.** **Total heating surface** $= D^2 \times L_1 \times \frac{6{\cdot}2}{d + 1{\cdot}5}$ sq. ft.

$L_1$ is the length of tubes in feet, and $d$ their external diameter in inches. In case of a double-ended boiler, $L_1$ is twice the length of the tubes. A boiler may have 5 to 8 per cent. more surface than given by this rule when the steam space is restricted.

**The weight of a cylindrical marine boiler** can be ascertained with fair accuracy by the following:—

**Rule 246.** **Weight of boiler** $= \frac{D^2 \times L \times \sqrt{p}}{C}$ tons.

$p$ is the pressure for which it is designed.

C is a factor, which, for single-ended boilers, is 768.

For double-ended boilers, with their combustion chambers common to opposite furnaces, $C = 733$.

For double-ended boilers, with separate chambers, $C = 800$.

Table XCIV.—Thermal Efficiency of various Boilers. Admiralty Boiler Committee, 1904.

| Name of Ship. | Type of Boiler. | Duration of Trial. | Rate of Combustion. | Thermal Efficiency. | Remarks. |
|---|---|---|---|---|---|
| | | Hours. | Lbs. per sq. ft. of Grate. | Per cent. | |
| Hermes, . | Babcock & Wilcox, . . | 30 | 20 | 81·0 | Vertical baffles ; air over fires. |
| ,, . | ,, ,, . . | 29 | 27 | 77·8 | ,, ,, ,, ,, |
| ,, . | ,, ,, . . | 12 | 20·5 | 75·8 | ,, ,, no air ,, |
| ,, . | ,, ,, . . | 24 | 19 | 71·0 | ,, ,, ,, ,, |
| ,, . | ,, ,, . . | 30 | 27 | 70·3 | ,, ,, ,, ,, |
| Medea, . . | Yarrow (large tube), . . | 26 | 18 | 75·7 | |
| ,, . . | ,, ,, . . | 8 | 40 | 69·5 | |
| ,, . . | ,, ,, . . | 24 | 17 | 75·0 | |
| ,, . . | ,, ,, . . | 24 | 21 | 75·0 | |
| Hyacinth, . | Belleville, . . . . | 24·5 | 16 | 77·2 | |
| ,, . | ,, . . . . | 11 | 20 | 73·3 | |
| ,, . | ,, . . . . | 8 | 27 | 65·0 | |
| Medusa, . | Dürr, . . . . | 8 | 35 | 64·8 | |
| ,, . | ,, . . . . | 26 | 16 | 63·8 | |
| ,, . | ,, . . . . | 24 | 18 | 61·7 | |
| ,, . | ,, . . . . | 24 | 21 | 60·3 | |
| Saxonia, . | Cylindrical (Howden f. d.), | 13 | 20 | 82·3 | |
| Minerva, . | ,, (Admiralty f. d.), | 25 | 14 | 69·7 | |
| ,, . | ,, | 8·5 | 29 | 68·4 | With retarders. |
| Sheldrake, . | Babcock & Wilcox, . | 12 | 18 | 66 | 1898. |
| Espiegle, . | ,, ,, . . | 9 | 13 | 73·2 | 1901. |
| Seagull, . | Niclausse, . . . . | 8 | 13 | 66·9 | Maximum efficiency. |
| Fantôme, . | ,, . . . . | 9 | 14 | 69·8 | ,, ,, |

Table XCV.—Particulars of Destroyers built by Messrs Yarrow & Co. Ltd.

| | H.M.S. "Hornet." | "Sokol." | "Mode." | H.M.S. 'Welland." | H.M.S. "T.B.D." |
|---|---|---|---|---|---|
| Length between perpendiculars, . . . | 180 ft. | 190 ft. | 220 ft. | 225 ft. | 270 ft. |
| Breadth, extreme, . | 18 ft. 6 ins. | 18 ft. 6 ins. | 20 ft. 6 ins. | 23 ft. 6 ins. | 26 ft. |
| Displacement, . tons | 220 | 179 | 330 | 585 | 860 |
| Engines, Kind of, {Twin screw | {3 cylinder triples | 3 cylinder triples | 4 cylinder triples. Balanced Y.S.T. system | 4 cylinder triples. Balanced Y.S.T. system | Turbines |
| Cylinders, Diameter and Stroke, . | $\frac{18-26-39\frac{1}{2}}{18}$ | $\frac{18-26-39\frac{1}{2}}{18}$ | $\frac{20\frac{1}{2}-31\frac{1}{2}-34-34}{18}$ | $\frac{23-35-38-38}{20}$ | Compd. turbines |
| Boilers, Kind of, and No., . . . | Yarrow straight tube 8 | Yarrow straight tube 8 | Yarrow straight tube 4 | Yarrow straight tube 4 | Yarrow straight tube 2 |
| Steam pressure, . . | 170 | 160 | 241 | 232 | Oil fuel |
| Grate area, square feet, | 172 | 172 | 220 | 270 | ... |
| Total heating surface, sq. feet, . . . | 8320 | 8320 | 13,000 | 16,000 | 16,600 |
| Air pressure, . . | $1\frac{1}{2}''$ | $1\frac{5}{16}''$ | 1·4" | 1·7" | ... |
| Coal per sq. ft. grate, per hour, . . lbs. | 55·34 | 45·0 | 68·0 | 47·0 | ... |
| Coal per I.H.P., . lbs. | 2·45 | 2·1 | ... | 1·63 | Oil 1·033 |
| Revolutions per minute, | 392·65 | 405 | 421·4 | 342·3 | ... |
| Mean total I.H.P. at full speed, . . | 3884 | 3800 | ... | 7766 | 17,900 |
| Speed, mean, knots | {27·628 for 3 hours | 29·762 for 3 hours | 32·428 for 3 hours | 26·244 for 4 hours | 35 |

Table XCVI.—Comparison of Water Tube Boilers with Others in various Ships (Trial Trips).

| Class of Ship. | Battleships. | | First-Class Cruisers. | | Second-Class Cruisers. | | | Third-Class Cruisers. | | Torpedo Gunboats. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Name. | Mean of Six. | Mean of Six. | "Edgar" Class. | "Diadem" Class. | "Diana" Class. | "Apollo" Class. | "Arrogant" Class. | "Pearl" Class. | "Pelorus" Class. | "Alarm" Class. | "Speedy." | "Sheldrake." | "Seagull." |
| Indicated horse-power, . | 12,414 | 13,500 | 12,851 | 16,961 | 9,846 | 9,274 | 10,240 | 7,469 | 7,152 | 3,665 | 4,703 | 4,050 | 3,546 |
| Boiler pressure, . . | 149 | 289 | 146·2 | 286·5 | 151 | 155 | 265 | 155 | 253 | 155 | .. | 151 | 145 |
| Air pressure in stokehole, . | 0·85 | 0·19 | 0·7 | 0·305 | 1·06 | 0·9 | 0·06 | 1·4 | 2·9 | 2·20 | 1·70 | 0·5 | 0·16 |
| Type of boiler, . . | Cylindrical. | Belleville. | Cylindrical. | Belleville. | Cylind. S.E. | Cylind. D.E. | Belleville, old. | Cylind. D.E. | Express. | Locomotive | Thornycroft. | Babcock. | Niclausse. |
| Grate area, . . sq. ft. | 817 | 1,050 | 812 | 1,460 | 624 | 575 | 869 | 410 | 353 | 171 | 204 | 252 | 276 |
| Total heating surface, ,, | 25,233 | 33,700 | 24,908 | 40,990 | 18,579 | 15,641 | 25,606 | 11,025 | 20,508 | 6,204 | 17,700 | 9,103 | 7,932 |
| Total weight of boilers, &c., . } tons | 724 | 623 | 646 | 767 | 548 | 434 | 463 | 314 | 172 | 124·4 | 132 | 124·8 | 134·7 |
| Ratio heating surface to grate, . . . . | 30·8 | 32·1 | 30·7 | 28·1 | 29·8 | 27·2 | 29·4 | 26·9 | 58·0 | 36·3 | 86·8 | 37·4 | 28·8 |
| Coal per I.H.P. per hr., lbs. | 2·29 | 1·72 | 1·70(*a*) | 1·85 | 2·18(*b*) | .. | 2·00 (*c*) | .. | 2·36(*d*) | .. | .. | 1·57 | 2·11 |
| ,, sq. ft. grate per hour, . } ,, | 34·8 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 25·6 | 27·1 |
| Weight of boilers, &c., per I.H.P., . } ,, | 131 | 103 | 111 | 101 | 124 | 104 | 99 | 98·6 | 52·0 | 76·0 | 62·9 | 69·9 | 85·1 |
| Weight of boilers, &c., per sq. ft. of grate, } ,, | 1,971 | 1,329 | 1,687 | 1,172 | 1,959 | 1,674 | 1,170 | 1,715 | 1,092 | 1,630 | 1,450 | 1,109 | 1,093 |
| I.H.P. per sq. ft. of grate, . | 15·2 | 12·9 | 15·2 | 11·6 | 15·79 | 16·1 | 11·82 | 18·2 | 20·9 | 21·4 | 23·0 | 16·0 | 12·8 |
| ,, ton of boiler, . | 17·1 | 21·7 | 20·1 | 22·1 | 17·97 | 20·7 | 22·5 | 23·8 | 43·1 | 29·5 | 39·6 | 32·5 | 26·3 |
| Total heating surface per I.H.P., . . . | 2·0 | 2·5 | 1·9 | 2·41 | 1·88 | 1·65 | 2·49 | 1·47 | 2·87 | 1·70 | 3·7 | 2·3 | 2·24 |
| Total weight of machinery, . . } tons | 1,341 | 1,290 | 1,161 | 1,540 | 915 | 740 | 825 | 539 | 383 | 211 | 212 | 205 | 214 |
| I.H.P. per ton of machinery, | 9·3 | 10·5 | 11·1 | 11·0 | 10·76 | 12·5 | 12·34 | 13·8 | 18·92 | 17·3 | 22·1 | 19·3 | 16·9 |
| Piston speed, . . ft. | 875 | 918 | 923 | 923 | 937 | 910 | 906 | 880 | 981 | 875 | 875 | 875 | .. |
| Coal consumption per I.H.P. at sea trial, . | 2·0 | 2·36 | 2·0 | 2·33 | 2·11 | .. | 2·36 | .. | .. | .. | .. | 2·14 | 2·15 |
| I.H.P. on sea trial, 1 ship, . | 8,029 | 8,163 | 8,824 | 13,517 | 6,694 | .. | 6,273 | .. | .. | .. | .. | 2,775 | 2,773 |

(*a*) At 10,517 I.H.P., (*b*) at 8307 I.H.P., (*c*) at 7275 H.P., (*d*) at 5388 H.P.

## EVAPORATION, HEATING SURFACE, &c.

The efficiency of the heating surface of a boiler depends on its position, the material and thickness of the plates, tubes, etc., the condition of the surfaces, and the circulation of the gases on the one side, and water in contact with them on the other, and their difference in temperature.

(1) **Position.**—The most efficient heating surface is that of the furnace crowns and combustion chambers; its high efficiency is due partly to the great difference of temperature between the two sides of the plates, partly to the freedom of the surfaces from deposit of soot or ash, and also in some measure to the fact that mineral or earthy matter deposited from the water does not readily adhere to vertical surfaces or to surfaces over which there is a good circulation of water.

Next in order of efficiency come the upper surfaces of the tubes, more especially of the *upper* tubes at the ends next the combustion chambers, for the reason that the flame and heated gases always seek the highest possible course and rapidly lose heat in passing through the tubes, entering them at, say, 2400° and leaving them under 800°, and also because the bubbles of steam disengage themselves more easily when they can rise directly from a surface.

When a "nest" of tubes is made very deep (*i.e.* when there are many rows one above another) it very often happens, for the reason just given, that the lower rows do little or no work, and might be often better absent from the boiler.

The surfaces below the level of the fire-bars, and the front tube-plates are generally left out of consideration as of no practical value.

(2) **Material.**—Amongst ordinary metals suitable for boiler construction, copper is by far the most efficient conductor of heat, its conductivity compared with iron or steel being about as 3 to 1; for various reasons, however, it is now very rarely used, and even brass boiler tubes are now hardly ever used. Steel for the shells, furnaces, and combustion chambers and for the tubes of water tube boilers, and iron and steel for the tubes of cylindrical boilers, are now the universal materials.

The high conductivity of copper so reduces the possible thermal difference between the fire side and the water side of a plate as to very materially diminish the local racking strains ordinarily produced during the operation of getting up steam, and for this reason it was peculiarly suitable for the fire boxes of torpedo-boat or other boilers in which the changes of temperature are sudden and great.

(3) **Condition of surfaces.**—A perfectly smooth and clean metallic surface does *not* give satisfactory results as a heating surface, as it is apt to cause sudden and violent intermittent ebullition, and, for this reason, few boilers work quite satisfactorily until they have become slightly corroded or have acquired a thin film of scale or deposit.

Since the cooling of the products of combustion takes place mainly in the tubes, soot is principally deposited in them, and they must be swept more or less frequently, according to the class of coal used, if the efficiency of the surface is to be maintained.

The external surfaces of the tubes, especially on the upper sides, are peculiarly liable to the accumulation of deposit where water containing salt or other mineral matter is used, as there is rapid evaporation, and the upward circulation is naturally sluggish and much impeded in large "nests" of tubes.

(4) **Circulation.**—The proper circulation of the water over the heating surfaces is of the very greatest importance, for no boiler in which the design and arrangements are such as to prevent such a circulation will ever steam satisfactorily or evaporate the quantity of water that it could do otherwise. The principal evils that result from defective circulation are deposit of scale on the evaporating surfaces, with the consequent overheating and buckling of plates and leaking of tubes, and the irregular and much diminished evaporation, accompanied by intermittent fits of "priming."

Special difficulty was found in getting a proper circulation of water over the tube plates next the fires of Naval boilers, mainly on account of the close pitch of the tubes and consequent contraction of the waterways; and similar difficulty was also experienced with the large flat (almost square) tops of the fire-boxes in boilers of the locomotive type.

In this latter case, the cure was found by fitting one or two "Galloway" tubes, a cure as effectual as was found many years ago to be the case in large double-ended cylindrical boilers. Better results, however, were obtained by dividing the fire-box completely by means of a water space, thus making two distinct furnaces.

**Diameter of tubes.**—The diameter of tube has a considerable influence on the efficiency of its surface, inasmuch as the contents increase as the square, whilst the surface increases as the diameter only. Thus if a 4-inch tube be substituted for two 2-inch ones, the absorbing surface will remain the same, but twice the quantity of gas will be passed, if the velocity of flow remain the same. If the velocity of flow be reduced in the 4-inch tube (to one half) so as to pass only the same quantity of gas as the two 2-inch ones, the 4-inch tube is still at a disadvantage,—inasmuch as the mean distance of the molecules of gas from its surface is greater than in the 2-inch ones. Howden's retarders somewhat remedied this defect.

When artificial or forced draught is used, the diameter of tube should be smaller than for natural draught,—if length of tube and velocity of gas are to remain the same,—since the gases have a higher temperature and must be divided into smaller threads or streams to prevent escape into the chimney at a wastefully high temperature: the draught, being positive and sharp, may be relied on to overcome the extra friction as well as to prevent the deposit of soot.

The diameters of tube in ordinary use are as follows :—

| | | | |
|---|---|---|---|
| (*a*) | Merchant steamer. | Chimney draught. | 3 inch to 3½ inch. |
| (*b*) | ,, ,, | Artificial ,, | 2½ inch to 2¾ inch. |
| (*c*) | Cruiser or battleship. | ,, ,, | 2½ inch, cyl. boilers. |

**Quantity of water evaporated per pound of coal.**—This of course depends not only on the quality of the coal, but on the type of boiler, the strength of draught, and the skill of the stoker. The average evaporation of water that may be expected with good fuel, under the various conditions indicated, is shown in Tables LXXXIX., XCIII., &c. A good cylindrical boiler should have an efficiency of 80 per cent. at least.

**Equivalent evaporation from and at 212° F.**—In order to compare evaporative results obtained with different temperatures of feed and pressures of steam, it is necessary to eliminate the effects of the varying conditions by reducing all the results to one common standard.

The standard generally employed is "the equivalent evaporation from and at 212° F.,"—*i.e.* the number of pounds of water that would be evaporated in each case, per pound of coal per hour, if the feed water were supplied at 212° F., and completely evaporated under the pressure (one atmosphere) due to that temperature.

On reference to the Table "Properties of Saturated Steam," it is seen that 966 thermal units are required to evaporate 1 lb. of water from 212°, and under a pressure of one atmosphere; so that, if the number of thermal units imparted to the water by each pound of coal during the test be determined, and divided by 966, the quotient will be the equivalent evaporation from and at 212°. The quantity of heat imparted by each pound of coal is determined by referring to column 7 of the same Table, opposite to the proper pressure, and there reading off the total heat, from 32°, contained in each pound of steam, deducting from this the number of units, above 32°, contained in each pound of feed-water as supplied, and then multiplying the remainder by the number of pounds of water evaporated, per pound of coal per hour, during the test. This operation is concisely expressed by the formula,—

$$N_{212}=N_T\left\{\frac{H-(t-32)}{966}\right\} \text{ (\textit{vide} Table C., p. 337)} \cdot$$

where $N_{212}$ = Number of lbs. of water evaporated per lb. coal, from and at 212°.

$N_T$ = Number of lbs. water evaporated per lb. coal under test conditions.

$H$ = total heat, from 32°, per lb. of steam at test temperature and pressure.

$t$ = temperature at which feed was supplied during test.

## Table XCVII.—Leading Particulars of some Boilers made since 1895.

| Designation. | Made in Accordance with the Rules of | Working Pressure. | Shell. | | | Furnaces. | | | Number of Combustion Chambers. | Tubes. | | | | Total Heating Surface. | Weight of | | | Weight per 100 feet of Total Heating Surface. | | | Sq. Ft. Total H.S. per Ton of Boiler. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Diameter. | Length. | Thickness. | Number. | Diameter. | Thickness. | | Number. | Diameter. | Length. | Surface. | | Boiler. | Water. | Total. | Boiler. | Water. | Total. | |
| | | Lbs. | Ft. In. | Ft. In. | In. | | In. | In. | | | In. | Ft. In. | Sq. Ft. | Sq. Ft. | Tons. | Tons. | Tons. | Tons. | Tons. | Tons. | |
| H D | Board of Trade, . | 190 | 16 5 | 17 10 | $1\frac{17}{32}$ | 8 | $40\frac{3}{4}$ | $\frac{19}{32}$ | ... | 848 | $2\frac{3}{4}$ | 7 2 | 4293 | 5112 | 83·0 | 47 | 130 | 1·62 | 0·92 | 2·54 | 61·6 |
| W F | Board of Trade and Lloyd's, | 180 | 16 4 | 17 6 | $1\frac{1}{2}$ | 8 | $40\frac{5}{8}$ | $\frac{9}{16}$ | ... | 840 | $2\frac{3}{4}$ | 7 $0\frac{1}{2}$ | 4170 | 4572 | 79·5 | 45·5 | 125 | 1·74 | 0·99 | 2·73 | 57·5 |
| P R | Board of Trade, . | 210 | 15 6 | 19 6 | $1\frac{13}{32}$ | 6 | 44 | $\frac{11}{16}$ | ... | 852 | $2\frac{5}{8}$ | 7 $10\frac{5}{8}$ | 4540 | 5210 | 83·0 | 46·75 | 129·7 | 1·59 | 0·91 | 2·50 | 62·7 |
| W N | ,, ,, . | 192 | 15 3 | 18 6 | $1\frac{15}{32}$ | 6 | $43\frac{1}{2}$ | $\frac{5}{8}$ | ... | 826 | $2\frac{5}{8}$ | 7 6 | 4180 | 4839 | 73·75 | 42·75 | 116·5 | 1·52 | 0·88 | 2·40 | 65·6 |
| B T | ,, ,, . | 214 | 14 5 | 19 10 | $1\frac{9}{16}$ | 6 | $40\frac{11}{16}$ | $\frac{21}{32}$ | ... | 740 | $2\frac{1}{2}$ | 7 11 | 3760 | 4415 | 75·5 | 41·75 | 117·25 | 1·71 | 0·95 | 2·66 | 58·5 |
| C A | Board of Trade and Lloyd's, | 200 | 14 3 | 19 6 | $1\frac{13}{32}$ | 6 | 42 | $\frac{5}{8}$ | 3 | 730 | $2\frac{1}{2}$ | 7 $7\frac{1}{2}$ | 3578 | 4575 | 68·5 | 40·5 | 109·0 | 1·50 | 0·88 | 2·38 | 66·8 |
| I D | Board of Trade, . | 216 | 14 2 | 19 6 | $1\frac{9}{16}$ | 6 | $40\frac{11}{16}$ | $\frac{11}{32}$ | ... | 612 | $2\frac{3}{4}$ | 8 0 | 3460 | 4071 | 72·5 | 40·5 | 113·0 | 1·78 | 1·00 | 2·78 | 56·1 |
| C P | Lloyd's, . . | 180 | 12 9 | 18 0 | $1\frac{3}{16}$ | 6 | 38 | $\frac{17}{32}$ | 3 | 616 | $2\frac{3}{4}$ | 6 6 | 2826 | 3415 | 41·0 | 30·1 | 71·1 | 1·20 | 0·88 | 2·08 | 83·3 |
| P E | Board of Trade and Lloyd's, | 190 | 11 6 | 18 9 | $1\frac{5}{32}$ | 4 | 42 | $\frac{19}{32}$ | 4 | 496 | $2\frac{3}{4}$ | 7 $1\frac{5}{8}$ | 2494 | 3000 | 43·2 | 22·5 | 65·7 | 1·44 | 0·75 | 2·19 | 69·4 |
| P G | Board of Trade and Lloyd's, | 180 | 15 0 | 11 9 | $1\frac{11}{32}$ | 3 | 45 | $\frac{19}{32}$ | 3 | 409 | $2\frac{3}{4}$ | 8 0 | 2355 | 2860 | 45·0 | 26·0 | 71·0 | 1·57 | 0·91 | 2·48 | 63·5 |
| O O | Board of Trade and Lloyd's, | 200 | 14 3 | 11 6 | $1\frac{5}{16}$ | 3 | 44 | $\frac{5}{8}$ | 3 | 238 | $3\frac{1}{2}$ | 7 9 | 1667 | 2138 | 40·6 | 19·0 | 59·6 | 1·90 | 0·89 | 2·79 | 52·6 |
| D N | Board of Trade, . | 160 | 14 3 | 11 6 | $1\frac{3}{16}$ | 3 | 44 | $\frac{17}{32}$ | 3 | 240 | $3\frac{1}{2}$ | 8 $1\frac{1}{2}$ | 1756 | 2232 | 37·85 | 19·55 | 57·4 | 1·70 | 0·88 | 2·58 | 56·3 |
| Z O | Board of Trade and Lloyd's, | 200 | 13 0 | 11 0 | $1\frac{5}{16}$ | 3 | 38 | $\frac{9}{16}$ | 3 | 234 | $3\frac{1}{4}$ | 7 6 | 1492 | 1907 | 34·8 | 17·2 | 52·0 | 1·82 | 0·90 | 2·72 | 54·8 |
| S A | Lloyd's, . . | 180 | 13 0 | 10 6 | $1\frac{1}{16}$ | 3 | 46 | $\frac{19}{32}$ | 3 | 176 | $3\frac{1}{2}$ | 6 9 | 1067 | 1335 | 26·5 | 16·5 | 43·0 | 1·92 | 1·24 | 3·16 | 50·4 |
| D O | Board of Trade and British Corporation, | 200 | 11 6 | 11 3 | $1\frac{1}{8}$ | 2 | 40 | $\frac{19}{32}$ | 2 | 172 | $3\frac{1}{4}$ | 7 7 | 1088 | 1327 | 27.3 | 13·8 | 41·1 | 1·91 | 1·04 | 2·95 | 50·0 serve tubes |
| L Z | Lloyd's, . . | 200 | 11 0 | 9 6 | 1 | 2 | $37\frac{3}{4}$ | $\frac{9}{16}$ | 2 | 132 | $3\frac{1}{4}$ | 6 3 | 686 | 880 | 18·85 | 10·5 | 29·35 | 2·14 | 1·19 | 3·33 | 46·7 |
| B Y | ,, . . | 160 | 11 0 | 9 6 | $\frac{15}{16}$ | 2 | 36 | $\frac{1}{2}$ | 2 | 146 | $3\frac{1}{4}$ | 6 3 | 761 | 1000 | 17·6 | 10·2 | 27·8 | 1·76 | 1·02 | 2·78 | 56·8 |
| D N | ,, . . | 200 | 10 0 | 9 6 | 1 | 2 | 35 | $\frac{5}{8}$ | 2 | 118 | $3\frac{1}{4}$ | 6 $5\frac{1}{2}$ | 650 | 820 | 17·3 | 8·75 | 26·05 | 2·11 | 1·67 | 3·78 | 47·4 |
| C O | ,, . . | 180 | 10 0 | 9 6 | $\frac{15}{16}$ | 2 | 35 | $\frac{9}{16}$ | 2 | 118 | $3\frac{1}{4}$ | 6 $5\frac{1}{2}$ | 650 | 820 | 16·4 | 8·75 | 25·15 | 2·00 | 1·67 | 3·67 | 50·0 |
| L N | Bureau Veritas, . | 180 | 8 9 | 9 9 | $\frac{27}{32}$ | 2 | 32 | $\frac{11}{16}$ | 1 | 82 | $3\frac{1}{4}$ | 6 9 | 561 | 600 | 13·4 | 7·45 | 20·85 | 2·23 | 1·24 | 3·47 | 44·8 |

## PROPORTIONS OF BOILERS.

**Grate surface.**—From the information given in Tables LXXXIX. and XCI., it will be an easy matter to determine the I.H.P. per foot of grate that may be anticipated (or the grate area that must be provided for any given I.H.P.) in the case of proposed new boilers of any of the types mentioned.

**Size of furnace.**—The furnaces of cylindrical boilers should not be less than 30 inches in diameter, nor more than 48 inches, unless in exceptional cases: wherever possible a diameter of not less than 40 inches should be given, as, with smaller furnaces,—owing to thickness of fire being practically constant for all diameters,—the space above the fuel is much contracted, and the combustion less perfect in consequence. As previously stated, long bars cannot be properly worked by hand, and are not so efficient as short ones: it is desirable that they should be limited to a length of 5 feet.

**Number of furnaces.**—

| | | | |
|---|---|---|---|
| Boilers up to | 9 feet diameter | may have | 1 furnace. |
| ,, ,, | 13 ft. 6 ins. | ,, | 2 furnaces. |
| ,, ,, | 15 feet | ,, | 3 ,, |
| ,, beyond | 15 feet | ,, | 4 ,, |

If the boilers are double-ended, the number of furnaces will, of course, be double the figure given above.

**The total heating surface** necessary and sufficient for each ship can be only determined by the special circumstances. It will be seen by reference to Table LXXXIX. that 10 to 13 lbs. of steam can be got per square foot of heating surface in water tube boilers, and as much as 9 lbs. from cylindrical; in both cases the draught is forced and the surfaces within and without quite clean but the efficiency is low for economic working. In a general way it may be taken that 8 lbs. per square foot of T.H. surface may be obtained economically with clean boilers; and to provide for dirty tubes, &c., 7 lbs. is as much as should be counted on with ships making a long enough voyage to cause fouling. For short voyages not exceeding eight hours and where economy of fuel is not of first importance, 8 may be taken. Assuming that the steam consumption on every kind of ship, including that of the engine's own auxiliaries (that is, air and circulating and feed pumps, etc.), is as follows:—

| | | |
|---|---|---|
| Turbines, best modern geared . | 10·5 | lbs. per H.P. hour. |
| Quadruples generally . . . | 14·0 | ,, ,, |
| Triples, best economic . . . | 14·7 | ,, ,, |
| ,, express service . . . | 15·5 | ,, ,, |
| Naval ships turbines (direct) . | 13·5 | ,, ,, |
| ,, reciprocators . . | 16·0 | ,, ,, |

Table XCVIII.—Particulars of Modern Cylindrical Boilers (Scantlings).

| | A. | B. | C. | D. | E. | F. | G. | H. |
|---|---|---|---|---|---|---|---|---|
| Diameter, mean, | 17′ 8″ | 15′ 3″ | 16′ 4″ | 16′ 2″ | 16′ 3″ | 15′ 6″ | 16′ 0″ | 15′ 3″ |
| Length over plates, | 22·0 | 18′ 6″ | 17′ 6″ | 11′ 0″ | 10′ 9″ | 11′ 0″ | 11′ 6″ | 12′ 0″ |
| Working pressure, lbs. | 195 | 230 | 180 | 220 | 190 | 215 | 180 | 180 |
| Rules on which designed, | B.T. | B.T. | B.T. B.V. | B.T. Ll. | Admlty. | B.T. Ll. | B.T. | B.T. |
| Number of furnaces, | 8 | 6 | 8 | 4 | 4 | 3 | 3 | 3 |
| Diameter of furnaces, ins. | 49·5 | $46\frac{1}{4}$ | $42\frac{1}{2}$ | $45\frac{1}{2}$ | $46\frac{3}{4}$ | $49\frac{1}{4}$ | 52 | 49 |
| Thickness of furnaces, ins. | 0·655 | $\frac{23}{32}$ | $\frac{18}{32}$ | $\frac{22}{32}$ | $\frac{19}{32}$ | $\frac{23}{32}$ | $\frac{21}{32}$ | $\frac{20}{32}$ |
| Number of combustion chambers, | 8 | 3 | 4 | 4 | 2 | 3 | 3 | 3 |
| Number of tubes, | 1064 | 644 | 840 | 438 | 578 | 396 | 342 | 463 |
| Diameter of tubes, ins. | $2\frac{1}{2}$ | $3\frac{1}{4}$ | $2\frac{3}{4}$ | $2\frac{3}{4}$ | $2\frac{1}{2}$ | $2\frac{3}{4}$ | $3\frac{1}{4}$ | $2\frac{1}{2}$ |
| Length of tubes, | 8′ $0\frac{1}{4}$″ | 7′ 4″ | 6′ 11″ | 7′ 8″ | 7′ $5\frac{1}{2}$″ | 7′ $2\frac{1}{4}$″ | 7′ 9″ | 7′ 6″ |
| Surface of tubes, sq. ft. | 5480 | 4017 | 4170 | 2417 | 2820 | 2044 | 2254 | 2272 |
| Surface of other parts, sq. ft. | 1120 | 669 | 861 | 544 | 467 | 462 | 516 | 465 |
| Total heating surface, sq. ft. | 6600 | 4686 | 5031 | 2961 | 3287 | 2506 | 2770 | 2737 |
| Grate area, sq. ft. | 169 | 121 | 160 | 90 | 94·2 | ... | 72 | 67·8 |
| Thickness of shell, ins. | 1·60 | $1\frac{7}{16}$ | $1\frac{1}{2}$ | $1\frac{21}{32}$ | $1\frac{17}{32}$ | $1\frac{21}{32}$ | $1\frac{7}{16}$ | $1\frac{11}{32}$ |
| Diameter of shell rivets, ins. | 1·63 | $1\frac{9}{16}$ | $1\frac{1}{2}$ | $1\frac{21}{32}$ | $1\frac{17}{32}$ | $1\frac{21}{32}$ | ... | $1\frac{7}{16}$ |
| Pitch of shell rivets, ins. | 10·5 | 9″ | 10″ | 10″ | 10″ | 10″ | ... | $9\frac{11}{16}$″ |
| Tensile test shell plates, | ... | 36–40 | 26 | 31–34 | 28–31 | 29–32 | 29–32 | 29–32 |
| Thickness of top end plates, | ... | $1\frac{1}{8}$ | $1\frac{1}{16}$ | $1\frac{1}{4}$ | $1\frac{1}{16}$ | $1\frac{1}{4}$ | $\frac{7}{8}$ | $\frac{7}{8}$ |
| ,, front tube plates, | 0·875 | 1 | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ |
| ,, back tube plates, | 0·812 | ... | ... | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ | $\frac{7}{8}$ |
| ,, combustion chambers, | 0·655 | $\frac{23}{32}$ | $\frac{19}{32}$ | $\frac{5}{8}$ | ... | ... | $\frac{11}{16}$ | $\frac{5}{8}$ |
| Weight as finished, tons | ... | ... | 78·5 | 57·0 | 52·0 | 53·0 | ... | ... |
| Weight per 100 sq. ft. T.H.S., tons | ... | ... | 1·560 | 1·926 | 1·582 | 2·115 | ... | ... |

To each case add the amount of steam required for other auxiliaries and domestic purposes, and the proper allowance of T.H. surface is obtained by dividing the sum by 7 or 8 as the case may be.

Of course each case may have a special rate of steam consumption of the main engines, and that should be taken instead of the average figures as above. The allowance for auxiliaries and domestic is also a variable quantity, but may be taken as 5 per cent. in an ordinary cargo steamer and may be as much as 15 per cent. on a passenger steamer in the North Atlantic or North Sea for winter service. With these assumptions the following holds good :—

## Table XCIX.—Total Heating Surface per I.H.P. of various Ships.

| Description of Ship and Service. | Steam Consumption. | | | Total Heating Surface per H.P. |
|---|---|---|---|---|
| | Main. | Auxly. | Total. | Sq. ft. |
| (1) Cargo steamer, general service, quadruple reciprocators . . | 14·0 | 0·70 | 14·70 | 2·100 |
| (2) Cargo steamer, general service, triple reciprocators . . . | 14·7 | 0·74 | 15·44 | 2·206 |
| (3) Passenger steamer, express long voyage, triple reciprocators . | 15·5 | 2·00 | 17·50 | 2·500 |
| (4) Passenger steamer, express short voyage, triple reciprocators . | 15·5 | 1·50 | 17·00 | 2·125 |
| (5) Passenger and cargo steamer, long voyage, quadruple reciprocators | 14·0 | 1·20 | 15·25 | 2·180 |
| (6) Atlantic passenger, express, turbine-driven . . . . . . | 12·5 | 2·50 | 15·00 | 2·143 |
| (7) Atlantic passenger and cargo, turbine-driven . . . . . | 12·5 | 1·50 | 14·00 | 2·000 |
| (8) Short service, express, cold climate, turbine-driven . . . . . | 13·0 | 1·65 | 14·65 | 1·83 |
| (9) Passenger and cargo steamer, geared, turbine-driven . . | 10·5 | 2·3 | 12·8 | 1·83 |
| (10) Naval ships, battleships, and large cruiser turbines . . . . . | 13·5 | 2·00 | 15·50 | 1·925 |
| (11) Naval ships, battleships, and large cruiser reciprocators . . . . | 16·0 | 2·00 | 18·00 | 2·25 |
| (12) Naval ships, scouts, and high-speed turbines . . . . . . | 14·0 | 2·00 | 16·00 | 2·00 |

Taking in each case the maximum average power developed by the engine during a run of not less than two hours as the basis, the allowance on the sea full power will be larger than above.

**Steam room.**—The steam room allowed in the various types of boiler, working at or about the pressures named, may be as follows:—

### Table XCIXa.—Allowance of Steam Room in Boilers (Cubic Feet per I.H.P.)

| Description of Boiler. | Type of Engine. | Working Pressure. | Allowance. |
|---|---|---|---|
| | | Lbs. | Cubic feet |
| (1) Cylindrical S. and D. ended, ordinary . | Triple and quadruple screw . | 200–230 | 0·36–0·32 |
| (2) Cylindrical S. and D. ended, ordinary . | Triple screw, slow revolutions . | 175–200 | 0·38–0·35 |
| (3) Cylindrical S. and D. ended, ordinary . | Triple screw, high revolutions . | 175–200 | 0·33–0·28 |
| (4) Cylindrical S. and D. ended, ordinary . | Compound paddle wheel . . | 120–140 | 0·43–0·38 |
| (5) Cylindrical S. and D. ended, ordinary . | Turbines . . | 160–180 | 0·24–0·18 |
| (6) Water tube as used in H.M. Navy . . | ,, . . | 180–200 | 0·085–·065 |
| (7) Water tube as used in H.M. Navy . . | Triple screw, very high revolutions | 210–250 | 0·095–·08 |

In cylindrical boilers the top row of tubes should be not less than ·28 × diameter of boiler from the top; if higher, the contraction of water surface is apt to cause priming.

Paddle and all slow-running engines should have considerably larger steam spaces than the faster running ones, but in practice weight of machinery is of such great importance that everything is cut down to the lowest limit; so, many compound paddle engines, working at pressures of 100 to 120 lbs., have not more than ·4 cubic foot per I.H.P.

**Water spaces, &c.**—The spaces between the furnaces themselves, between the furnaces and the shell, and between the combustion chambers, although sometimes made as narrow as 5 inches, are better 5½ inches, and even 6 inches when possible; the space between back of combustion chamber and end of boiler should be 6 inches at the bottom, increasing to 9, 10, or even 12 inches at the top.

A suitable pitch for tubes is 1·35 to 1·4 × external diameter of tube; where weight is of great importance, as in Naval vessels, the tubes are sometimes pitched a little closer than this, but with water spaces so contracted there is always risk of priming. A fair average pitch is given by the rule,—

**Rule 247.** **Pitch of tubes** $= D + \frac{\sqrt{D}}{2} + \cdot 35$ inch;

where D is outside diameter of tube in inches.

Whenever possible the tubes should be placed in horizontal and vertical rows, and not arranged in any diagonal lines or zigzag fashion.

The clear space between the nests of tubes should never be less than 10 inches, and when possible should be 10½ or 11 inches,—in order that a man may be able to get down: manholes between the furnaces and tubes can be then dispensed with—an important point,—for the life of a boiler depends to a considerable extent on the number of openings in its shell,—the leakage from such openings being most destructive. For the same reason, longitudinal seams below the water level should be avoided as much as possible.

Tables XCVII. and XCVIII. give particulars (dimensions and weights) of a number of boilers that have been actually constructed in accordance with different rules as indicated. The weights given are those of "bare boiler," without either furnace fittings or boiler mountings.

## Table C.—Multipliers for Converting Weight of Water actually evaporated to the Equivalent Quantity from and at 212° F.

| Temp. of Feed Water. | Boiler pressure, being absolute pressure minus 14·7 lbs. | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 110 | 120 | 130 | 140 | 150 | 160 | 170 | 180 | 190 | 200 | 210 | 220 | 230 | 240 | 250 |
| F° | | | | | | | | | | | | | | | |
| 60 | 1·200 | 1·202 | 1·203 | 1·205 | 1·207 | 1·208 | 1·210 | 1·211 | 1·212 | 1·214 | 1·215 | 1·216 | 1·217 | 1·218 | 1·219 |
| 70 | 1·189 | 1·191 | 1·193 | 1·194 | 1·196 | 1·197 | 1·199 | 1·200 | 1·202 | 1·203 | 1·205 | 1·206 | 1·207 | 1·208 | 1·209 |
| 80 | 1·179 | 1·181 | 1·183 | 1·184 | 1·186 | 1·187 | 1·189 | 1·190 | 1·192 | 1·193 | 1·194 | 1·195 | 1·196 | 1·198 | 1·199 |
| 90 | 1·169 | 1·170 | 1·172 | 1·174 | 1·176 | 1·177 | 1·179 | 1·180 | 1·181 | 1·183 | 1·184 | 1·185 | 1·186 | 1·187 | 1·188 |
| 100 | 1·158 | 1·160 | 1·162 | 1·164 | 1·165 | 1·167 | 1·168 | 1·170 | 1·171 | 1·172 | 1·174 | 1·175 | 1·176 | 1·177 | 1·178 |
| 110 | 1·148 | 1·150 | 1·152 | 1·153 | 1·155 | 1·156 | 1·158 | 1·159 | 1·160 | 1·162 | 1·163 | 1·164 | 1·166 | 1·167 | 1·168 |
| 120 | 1·138 | 1·140 | 1·141 | 1·143 | 1·145 | 1·146 | 1·147 | 1·149 | 1·150 | 1·151 | 1·153 | 1·154 | 1·155 | 1·156 | 1·157 |
| 130 | 1·127 | 1·129 | 1·130 | 1·132 | 1·134 | 1·136 | 1·137 | 1·138 | 1·140 | 1·141 | 1·142 | 1·144 | 1·145 | 1·146 | 1·147 |
| 140 | 1·117 | 1·119 | 1·120 | 1·122 | 1·124 | 1·125 | 1·127 | 1·128 | 1·129 | 1·131 | 1·132 | 1·133 | 1·134 | 1·135 | 1·136 |
| 150 | 1·106 | 1·108 | 1·110 | 1·111 | 1·113 | 1·115 | 1·116 | 1·118 | 1·119 | 1·120 | 1·121 | 1·123 | 1·124 | 1·125 | 1·126 |
| 160 | 1·096 | 1·098 | 1·100 | 1·101 | 1·103 | 1·104 | 1·106 | 1·107 | 1·108 | 1·110 | 1·111 | 1·112 | 1·113 | 1·115 | 1·116 |
| 170 | 1·085 | 1·087 | 1·089 | 1·091 | 1·092 | 1·094 | 1·095 | 1·097 | 1·098 | 1·099 | 1·101 | 1·102 | 1·103 | 1·104 | 1·105 |
| 180 | 1·075 | 1·077 | 1·079 | 1·080 | 1·082 | 1·083 | 1·085 | 1·086 | 1·088 | 1·089 | 1·090 | 1·091 | 1·092 | 1·094 | 1·095 |
| 190 | 1·065 | 1·066 | 1·068 | 1·070 | 1·071 | 1·073 | 1·074 | 1·076 | 1·077 | 1·078 | 1·080 | 1·081 | 1·082 | 1·083 | 1·084 |
| 200 | 1·054 | 1·056 | 1·058 | 1·059 | 1·061 | 1·063 | 1·064 | 1·065 | 1·067 | 1·068 | 1·069 | 1·071 | 1·072 | 1·073 | 1·074 |
| 210 | 1·044 | 1·046 | 1·047 | 1·049 | 1·051 | 1·052 | 1·053 | 1·055 | 1·056 | 1·057 | 1·059 | 1·060 | 1·061 | 1·062 | 1·063 |

Table CI.—Relative Weights of various Boiler Installations. Admiralty Boiler Committee, 1904.

| Name of Ship. | Type of Boiler. | Weight of Boiler Installation. | Maximum Output Steam per Hour. | Output per Hour per Ton of Boiler. | Remarks. |
|---|---|---|---|---|---|
| | | Tons. | Lbs. | Lbs. | |
| Saxonia, . | Cylindrical (Howden), . | 1000 | 132,600 | 132·6 | |
| Minerva, . | ,, (Admiralty f. d.), | 567 | 167,100 | 295 | With Retarders. |
| ,, | ,, ,, | 556 | 156,200 | 280 | Without ,, |
| Hyacinth, | Belleville, . . . . | 454 | 178,700 | 394 | |
| Medea, . . | Yarrow, . . . . | 330 | 157,800 | 478 | |
| Medusa, . | Dürr, . . . . | 314 | 158,000 | 503 | |
| Hermes, . | Babcock & Wilcox, . . | 490 | 200,000 | 410 | With baffles and air over fires. |
| ,, | ,, ,, . . | 481 | 182,300 | 380 | As originally fitted, 1903. |
| Sheldrake, | ,, ,, . . | 125 | 43,840 | 351 | 1898. |
| Espiegle, . | ,, ,, . . | 95 | 24,780 | 261 | 1901. |
| Seagull, . | Niclausse, . . . . | 135 | 28,450 | 359 | |
| Fantôme, . | ,, . . . . | 76·5 | 24,750 | 297 | |

Table CIa.—Comparison of Steel Boilers for 160 lbs. pressure by Board of Trade Rules (prior to 1895).

| General Description. | No. of Boilers. | Shell. | | Furnaces. | | | Tubes. | | | | Total Heating Surface. | Total Capacity. | Weight. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Diam. | Lgth. | No. | Diam. | Lgth. | No. | Diam. | Lgth. | Surface. | | | Boiler | Water | Total. |
| | | Ft. in. | Ft. in. | | Ins. | Ft. in. | | Ins. | Ft. in. | Sq. ft. | Sq. ft. | C. ft. | Tons. | Tons. | Tons. |
| Double-ended boiler, having two combustion chambers, each common to *opposite* furnaces, . . | 1 | 12 0 | 14 0 | 4 | 41 | 5 6 | 376 | 3¼ | 5 6 | 1715 | 2000 | 1582 | 35·3 | 21 | 56·3 |
| Double-ended boiler, having two combustion chambers, each common to furnaces at *same* end, . . | 1 | 12 0 | 14 0 | 4 | 41 | 5 3 | 376 | 3¼ | 5 3 | 1630 | 2000 | 1582 | 36 | 21 | 57 |
| Double-ended boiler, having four combustion chambers, one to each furnace, . . . . | 1 | 12 0 | 14 0 | 4 | 41 | 5 2 | 376 | 3¼ | 5 2 | 1610 | 2000 | 1582 | 36·4 | 21·5 | 57·9 |
| Single-ended boiler, having three combustion chambers, one to each furnace, | 1 | 14 6 | 9 7 | 3 | 45 | 6 9 | 274 | 3¼ | 9 | 1640 | 2000 | 1582 | 37·25 | 22 | 59·25 |
| Two single-ended boilers, each having two combustion chambers, one to each furnace, . . . | 2 | 11 0 | 8 4 | 4 | 39 | 5 10 | 340 | 3¼ | 5 10 | 1645 | 2000 | 1582 | 39·2 | 21·4 | 60·6 |

## STEEL BOILERS—CONSTRUCTION.

The following is a summary of the Admiralty, Board of Trade, and Lloyd's rules relating to the construction of steel boilers:—

**Admiralty tests of Boiler Steel.**—All steel to be made by the acid open-hearth process. Every plate, &c., used is to be tested, and must comply with the requirements stated below.

### Table CII.—Admiralty Tensile Tests.

| Description of Material. | Minimum ultimate tensile strength, tons per square inch. | Maximum ultimate tensile strength, tons per square inch. | Minimum elongation in 8 inches, per cent. |
|---|---|---|---|
| Not exposed to flame & not flanged, | 27 | 30 | 20% |
| Exposed to flame and flanged, . | 24 | 27 | 25% |
| Rivet bars, . . . . . . | 24 | 27 | 25% |
| Steam-pipe plates, . . . | 24 | 27 | 33% |
| Corrugated or ribbed furnace, . | 23 | 25 | 27% |
| Tube forgings (annealed), . . | 21 | 24 | 23% |
| Pieces cut from tubes (annealed), . | ... | 26 | 27% |

For bending tests the specimens are to be heated to a low cherry red, and then cooled in water at 82° F. Strips of plate 1½ inches wide must bend double in press,—inner radius being 1½ times thickness of plate. For pieces of rivet bar, inner radius to equal radius of bar; and for strips from tubes, ½ inch. Plates exposed to flame are also to be tested by welding and forging, some of the welds being broken in the testing machine to ascertain degree of perfection. Angle, tee, and bar steel is to stand such other forge tests as the overseer may direct.

Samples from each batch of rivets are also to stand the following tests:—To be bent double (cold), inner radius of bend being equal to radius of rivet; to be bent double (hot) and hammered till the two parts of shank meet; head to be flattened (hot) without cracking until its diameter is 2½ times diameter of shank; and shank to be nicked on one side, and bent over to show quality of metal.

Tubes under $\frac{3}{16}$ inch thick to stand drifting out at each end, when hot, to 20 per cent. larger diameter, and expanding out, when cold, to 12½ per cent. larger diameter, and thicker tubes to go half these amounts. Sections of thin tubes, 2 inches long, to bear hammering down, cold, to 1¼-inch if lap welded, and to 1 inch if solid drawn; and to bear flattening, if below $\frac{3}{16}$-inch thick, until sides touch, and if thicker until sides come within twice thickness, in all cases without fracture.

## Board of Trade Tests of Boiler and other Steel. Instructions to Surveyors.

### MANUFACTURE AND TESTING OF STEEL MATERIAL INTENDED FOR BOILERS AND MACHINERY UNDER BOARD OF TRADE SURVEY.

1. *Introduction.*—The Board having had under consideration the reports issued by the Engineering Standards Committee, prepared the following amended instructions for the guidance of their Surveyors, which came in force on 1st August 1908.

### GENERAL CONDITIONS.

109. *Process of manufacture: annealing shell plates.* — All steel intended for use in the construction of boilers and for forgings should be made by the open-hearth process. Boiler plates should be of acid quality; but the other portions of boilers, and forgings, may be made of either acid or basic steel. In the case of castings, the steel may be made by any process which has been approved by the Board of Trade.

It is very desirable that all plates (especially those of great thickness) intended for the shells of boilers should be annealed, but it is important that the process should be carefully effected, the plates being heated singly to a suitable temperature, in a properly constructed furnace, and allowed to cool separately and uniformly out of the furnace.

110. *Selection and treatment of test-pieces.* — All the test-pieces required should be selected by the Surveyor, and, except where otherwise specified, the tests should be made in his presence at the place of manufacture, and before the despatch of the material; and the stamping of test-pieces shall be done after all the heating or annealing is completed.

If any material is annealed or otherwise heat-treated, the test-pieces should be similarly and simultaneously treated with the material before they are tested. The specimens should not be further heated, excepting those for temper-bending tests, which should be heated to a blood red and quenched in water at a temperature not exceeding 80° F.

As regards forgings, the test-pieces should be taken from a part of the forgings of sectional dimensions not less than those of the body of the forging, and they should be machined to size without further forging down.

Test-pieces should not be cut off forgings or castings until they have been stamped by the Surveyor after the annealing has been completed.

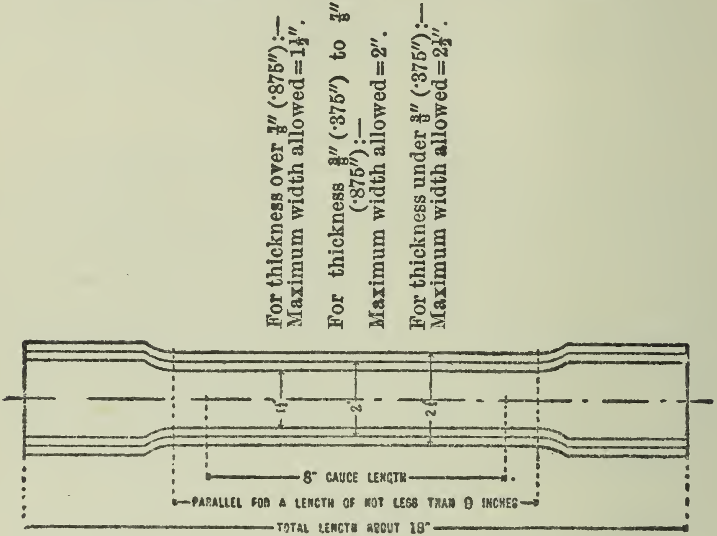
When a number of articles are cut from one plate, bar or forging, the number of tests required should be the same as that required from the original piece, provided the articles have not been further heated or forged, and can be identified as having formed part of the original piece.

When a number of small forgings are made from the same ingot, or a number of small castings from the same charge of steel, the full number of tests specified herein need not be made ; tensile and bending tests at the rate of one of each for every four articles will, as a rule, in such cases be sufficient.

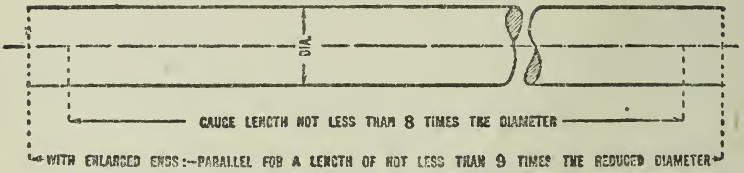
111. *Standard test-pieces.*—The forms and dimensions of test-pieces should be as follows :—

### (a) Tensile Tests.

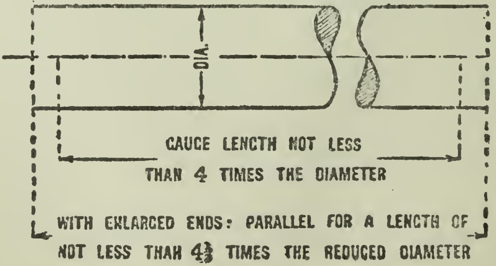
PLATES, TEE, AND ANGLE BARS—Test Piece A.


For thickness over $\frac{7}{8}''$ (·875"):—
Maximum width allowed = $1\frac{1}{2}''$.
For thickness $\frac{3}{8}''$ (·375") to $\frac{7}{8}''$ (·875"):—
Maximum width allowed = 2".
For thickness under $\frac{3}{8}''$ (·375"):—
Maximum width allowed = $2\frac{1}{2}''$.
$1\frac{1}{2}''$
2"
$2\frac{1}{2}''$
8" GAUGE LENGTH
PARALLEL FOR A LENGTH OF NOT LESS THAN 9 INCHES
TOTAL LENGTH ABOUT 18"

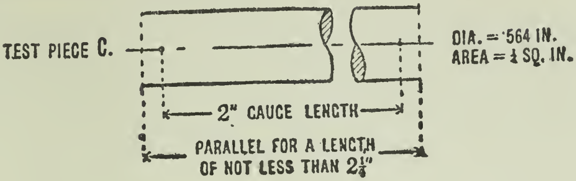
BARS, RODS, AND STAYS—Test Piece B.

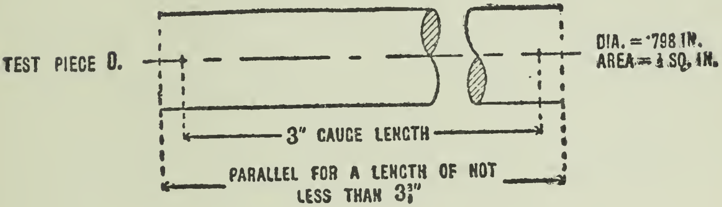

DIA.
GAUGE LENGTH NOT LESS THAN 8 TIMES THE DIAMETER
WITH ENLARGED ENDS:—PARALLEL FOR A LENGTH OF NOT LESS THAN 9 TIMES THE REDUCED DIAMETER

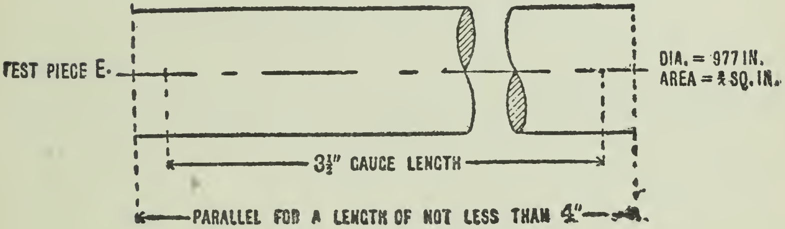
ALTERNATIVE TEST PIECE F—For Test Pieces over 1 inch diameter.


DIA.
GAUGE LENGTH NOT LESS THAN 4 TIMES THE DIAMETER
WITH ENLARGED ENDS: PARALLEL FOR A LENGTH OF NOT LESS THAN $4\frac{1}{2}$ TIMES THE REDUCED DIAMETER

FORGINGS AND CASTINGS—Test Pieces C, D and E.







The gauge length and the parallel portion of the above test-pieces should be as shown. The form of the ends to be as required in order to suit the various methods employed for gripping the test-pieces.

Any reduction of the specimens to the form required should be effected by machine, and, whenever practicable, the rolled surfaces should be retained on two opposite sides of the test-pieces taken from plates, angle bars, and tee bars.

### (b) Bending Tests.

The specimens sheared from plates, angle bars, and tee bars, for bending tests, shall not be less than 1½ inches wide; but for small bars the whole section may be used. The rough edge caused by shearing samples ½ inch in thickness, and above, may be removed by filing or grinding; and samples, 1 inch in thickness, and above, may have the edges machined. The bending specimens of round bars should, whenever practicable, be of the full diameter of the bars as rolled, but those of large section may be turned down to a diameter of 2 inches, if desired.

The bending tests of forgings and castings should be made with rectangular test-pieces, 1 inch wide by $\frac{3}{4}$ inch thick, which should be machined to size and have the corners rounded to a radius of $\frac{1}{16}$ inch: they should be bent over their thinner section.

112. *Duplicate tests.*—Should either a tensile or a bend test fail to fulfil the test requirements, and the Surveyor considers that the test-piece does not fairly represent the quality of the material, two duplicate specimens may, if the maker wishes, be tested; and if the results obtained from both are satisfactory, the quality of the article shall be judged therefrom and not from the original test which failed. If, however, either of the duplicate tests fails, the article or articles represented shall be rejected.

Should any tensile test-piece break at a point outside the middle half of its gauge length, the test may, at the makers' option and with the Surveyor's approval, be discarded, and another test may be made from the same plate, bar, forging, or casting.

113. *Stamping of material.*—Every article shall be stamped with a number or identification mark such that the charge of steel from which it was made can be readily identified. In addition to this, plates and bars shall be stamped with the maker's name or trade mark, and plates with the results of any tests which are made from them.

114. *Reporting test results.*—After witnessing the steel tests, the Surveyors should, except in the cases subsequently mentioned, submit the form Surveys 24 recording the results as soon after the tests are made as is convenient, without waiting for the completion of the order; and should be forwarded without loss of time to the Surveyors at the port at which the material is to be used.

115. *Freedom from defects, &c.*—The finished material should be sound and free from cracks, surface flaws, and laminations, and no hammer-dressing, patching, burning, or electric welding is permissible.

Local dressing of shell plates is undesirable, but if in any case the steel-makers wish to adopt this method of removing surface defects, the attention of the Surveyor must first be called to the defect, and no dressing must be done until he has inspected the part and satisfied himself that no crack exists. He should then stamp the adjacent part with his initials and record what he has done on the form Surveys 24, which should be forwarded to the Engineer Surveyor-in-Chief for transmission to the Surveyor who will inspect the boiler for which the plate is intended. In no circumstances whatever is hammer-dressing allowed, and the means for removing a surface defect must be confined to chipping and filing.

When the Surveyor who inspects the boiler during construction receives the form Surveys 24, he should, if practicable, arrange for the part to which attention has been drawn to be kept on the outside of the boiler, and he must specially examine the part after it has been rolled to the cylindrical form.

It is most necessary that the Surveyor should carefully scrutinise the inner and outer surfaces of all cyclindrical shell plates, with a view to detecting cracks, while the plates are being worked in the boilers. If he has any doubt about a part, a light chipping should be taken off the surface, in order to see if the chipping divides at a crack.

Boiler-makers should be requested to examine carefully all shell plates in the various stages of working, as they have the best opportunity of discovering defects, and occasionally cracks develop when working the plates in the boiler shop; but such inspection does not in any way relieve the Surveyor from his duty personally to inspect the plates.

Every precaution should be taken by the steel-makers to prevent a defective plate leaving their works, as they are responsible for supplying sound material. That a great deal can be done in this respect by the makers is clear from the fact that there are some steel-makers manufacturing a large amount of plating with regard to which the Board have no record of a defective shell plate.

In the event of any material proving unsatisfactory in the course of working or machining, it should be rejected notwithstanding any previous certification of satisfactory testing.

## STEEL FOR USE IN BOILERS.

116. *General instructions.*—The following instructions regarding boiler material refer to steel of ordinary mild quality. Where high tensile steel is used, the requirements specified by the Board in each case should be adhered to.

### *Plates.*

117. *Number and nature of tests.*—A tensile and a bending test should be taken from each plate, as rolled; but, when the weight of the plate exceeds two and a half tons, a tensile and a bending test should be taken from each end. Bending tests only, however, need be made from plates for which a greater stress than is allowed for iron is not desired.

The plates for man-hole doors, and for compensating rings around the openings for doors, should be tested in the usual manner.

118. *Tensile strength and elongation.*—The tensile strength of plates not intended to be worked in the fire or exposed to flame, for which special limits have not been sanctioned, should be between 27 and 32 tons per square inch: that of other plates, from 26 to 30 tons per square inch. The elongation should not be less than 20 per cent. in a length of 8 inches for material $\frac{3}{8}$ inch in thickness and upwards which is required to have a tensile strength of 27 to 32 tons per square inch, and not less than 23 per cent. if the tensile strength is required to be between 26 and 30 tons per square inch. For material under $\frac{3}{8}$ inch in thickness, the elongation may be reduced; but, for each eighth of an inch of diminution in thickness, the reduction should not be more than 3 per cent. below the elongations mentioned.

119. *Bend tests.*—Bending test-pieces should withstand being bent,

without fracture, until the sides are parallel at a distance apart of not more than three times the thickness of the specimen. The bending tests of plates not intended to be worked in the fire or exposed to flame may be made with strips in the same condition as the plates: those from other plates should be made with strips which have been tempered.

120. *Witnessing of tests by Surveyor.*—It is very desirable that the Surveyor should witness the whole of these tests; but, in the case of plates made from steel manufactured by any of the makers whose names are given in section 108, he need only select and witness tests from one in four of the plates of each thickness, unless the weight of the plate is over two and a half tons, or special limits of strength, or, in the case of shell plates, a minimum tensile strength exceeding 27 tons is required, in which cases the Surveyor should see the tests made from all the plates.

### *Angle, Rivet, and Stay Bars.*

121. *Number and nature of tests.*—One tensile test should be made from each 15, or part of 15, bars rolled of each section or diameter from the same charge, but not less than two tensile tests should be made, unless the total number of bars rolled from the same charge is 8, or less than 8, and the bars are of the same section or diameter, when one test will suffice. For round bars 1¾ inches in diameter, and under, the numbers 50 and 20 should be substituted for 15 and 8 respectively, as determining the number of tests necessary.

A cold and a temper bend should be made from stay bars in the same proportion as that in which tensile tests are required; and a cold or temper bend should be made from each angle or tee bar rolled. No bending tests need be made from rivet bars.

122. *Tensile strength and elongation of stays, angles, and tee bars.*—The tensile strength of longitudinal stays, angles, and tee bars should be between 27 and 32 tons per square inch, with an elongation of not less than 20 per cent. measured on the appropriate standard test-piece (A or B). For bars for combustion-chamber stays, the tensile strength should be between 26 and 32 tons per square inch, with an elongation of not less than 23 per cent. measured on the standard test-piece. When, however, stay bars are tested on a gauge length of four times the diameter (test-piece F), the elongations should be 24 per cent. and 28 per cent. respectively.

For tee or angle bars under ⅜ inch in thickness, the elongation may be 3 per cent. below that specified for plates.

123. *Bend tests.*—Bending test-pieces should withstand being bent, without fracture, until the sides are parallel at a distance apart of not more than three times the thickness or diameter of the specimen.

124. *Rivet bars.*—The tensile strength of rivet bars should be between 26 and 30 tons per square inch, with an elongation of not less than 25 per cent. measured on the standard test-piece B, or 30 per cent. if measured on test-piece F.

### *Rivets.*

125. *Nature of tests.*—A few rivets of each size should be selected by the Surveyor from the bulk, and should be subjected to the following tests:—

(*a*) The rivet shanks to be bent cold and hammered until the two parts of the shank touch, without fracture on the outside of the bend.

(*b*) The rivet heads to be flattened, while hot, until their diameter is two and a half times the diameter of the shank, without cracking at the edges.

A few check tensile tests of rivets should also be made when the Surveyor considers it necessary. The elongation should, when practicable, be taken in a length of two and a half times the diameter of the prepared part; the tensile strength should be from 27 to 32 tons per square inch and the contraction of area about 60 per cent.

### STEEL FORGINGS.

131. *General.*—The forgings shall be made from sound ingots, and not more than the lower two-thirds of the ingot may be utilised for forging. The sectional area of the body of the forging may not exceed one-fifth of the sectional area of the original ingot; and no part of the forging shall have more than two-thirds of the sectional area of the ingot. All ingot steel forgings shall, after completion, be thoroughly annealed at a uniform temperature; and if any subsequent heating is done, the forging shall, if required by the Surveyor, be again annealed.

128. *Number of tests.*—At least one tensile and one bend test shall be taken from each forging; but if the weight exceeds three tons, a tensile and a bending test shall be taken from each end.

129. *Tensile strength and elongation.*—The tensile strength of steel forgings shall not exceed 40 tons per square inch; and the elongation, measured on the appropriate standard test-piece C, D, or E, shall not be less than 17 per cent. for 40-ton steel; and in no case may the sum of the tensile strength and the corresponding elongation be less than 57.

130. *Bend tests.*—The bending test-pieces should withstand being bent through an angle of 180° without fracture; the internal radius of the bend being not greater than that specified below:—

| Maximum specified tensile strength of forging. | Internal radius of test-piece after bending. |
|---|---|
| | inch. |
| Up to 32 tons per square inch ... ... ... | $\frac{1}{4}$ |
| Above 32 tons and up to 36 tons per square inch | $\frac{3}{8}$ |
| ,, 36 ,, 40 ,, ,, | $\frac{5}{8}$ |

## STEEL CASTINGS.

135. *General.*—All steel castings shall be thoroughly annealed at a uniform temperature and shall be allowed to cool down prior to removal from the annealing furnace; and if subsequently heated, with the Surveyor's approval, shall again be similarly annealed, if required by the Surveyor.

132. *Number and nature of tests.*—No tests need be made from unimportant steel castings or from steel castings which are used for articles usually made of cast-iron, if the scantlings are not materially reduced below what would be required if cast iron were used. All other steel castings shall be tested as follows:—

At least one tensile and one bending test should be made from the castings from each charge; and where a casting is made from more than one charge, at least four tensile and four bending tests should be made from pieces cast as far apart as possible on the casting and as near the top and the bottom respectively as practicable.

When more than one casting is made from one charge, at least one tensile and one bending test should be made from the castings run from one common pouring head; but separate tests should be made from each casting or set of castings run from each separate pouring head. Small castings may, however, be dealt with in accordance with the provisions of section 110.

133. *Tensile strength and elongation.*—The tension strength may range from 26 to 40 tons per square inch, with an elongation, measured on the standard test-piece C, D, or E, of not less than 15 per cent. If, however, the castings are to be used for the more important pieces of machinery, such as pistons, etc., or for articles usually made of wrought material, the elongation should not be less than 20 per cent. where the corresponding tensile strength is between 26 and 35 tons per square inch.

134. *Bend tests.*—The bending test-pieces must withstand being bent, without fracture, through an angle of 60° if the tensile strength is between 35 and 40 tons per square inch; and in the case of other castings, through an angle of 90°. But if they are required to be of the superior quality referred to above, the angle must not be less than 120°.

## TUBES.

126. *Number and Nature of Tests.*—

### (a) *Solid-drawn Steel Steam Pipes, Boiler Tubes, &c., subject to Internal Pressure.*

The makers should take a few samples from each batch of tubes and test them for tensile strength and elongation. A bending test should also be made by them from the scrap end of each tube drawn.

Tensile and bending tests should also be made in the Surveyor's presence from specimens selected by him in the following proportion, from the tubes made from *each charge*:—

Tubes up to and including 3 inches in diameter : 1 in 40 or part thereof.

Tubes above 3 inches up to and including 4 inches in diameter : 1 in 20 or part thereof.

Tubes above 4 inches up to and including 5 inches in diameter : 1 in 10 or part thereof.

Tubes above 5 inches up to and including 7 inches in diameter : 1 in 6 or part thereof.

Tubes above 7 inches in diameter : 1 in 4 or part thereof.

The tensile strength should range between 23 and 30 tons per square inch, and the elongation should not be less than 20 per cent. in a length of 8 inches, or 18 per cent. if the thickness of the tubes is less than ¼ of an inch.

All the tubes should be tested by the makers to a suitable hydraulic pressure, and the tests of at least 25 per cent. of them should be witnessed by the Surveyor. The tests of all steam pipes should, however, be witnessed by the Surveyor on completion of the pipes, that is, after they have been bent to shape and the flanges have been secured in position (*see* section 174).

### (b) *Solid-drawn Steel Tubes subject to External Pressure.*

If no allowance over that given for iron tubes is required, a few bending tests should be made from the scrap ends of the stay tubes, but special tests need not be made from the ordinary tubes if the Surveyor finds the general quality of the material satisfactory and he is satisfied.

If allowance over iron is required, tensile and bending tests should be witnessed by the Surveyor in the proportions given for solid-drawn steel steam pipes. The tensile strength should range between 23 and 30 tons per square inch, and the elongation should be at least that required for similar solid-drawn steam pipes.

All the tubes should be tested by the makers to a suitable hydraulic pressure, but the tests need not be witnessed by the Surveyor if he is satisfied that the tubes have been duly tested by the makers.

### (c) *Steel Lap-welded Tubes subject to External Pressure.*

(i) *Steel tubes for which no allowance over iron is required.*—A few bending tests should be made from the scrap ends of the stay tubes or the strips from which they are made, but special tests need not be made from the ordinary tubes if the general nature of the material has been found satisfactory and the Surveyor is satisfied.

(ii) *Steel stay tubes for which allowance over iron is required.*—Tensile and bending tests should be made from 25 per cent. of the *strips* from which the tubes are made. The tensile strength should range between 23 and 30 tons per square inch, and the elongation should be at least 20 per cent. in a length of 8 inches when the strips are tested in their normal condition.

All the tubes should be tested by the makers to a suitable hydraulic pressure, but the tests need not be witnessed by the Surveyor if he is satisfied that the tubes have been duly tested by the makers.

127. *General Requirements for all Tubes.*—The hydraulic test should not, in any case, be less than three times the working pressure, and it should not exceed four times the pressure given by the rule:—

$$\frac{6000 \times \text{thickness in inches}}{\text{inside diameter in inches}} = \text{pressure,}$$

in the case of lap-welded tubes, or five times that pressure in the case of solid-drawn steel tubes.

All the tests mentioned should be made in the Surveyor's presence, except where otherwise stated, and such means as may be necessary should be taken to satisfy the Surveyor that the specimens he may have to test have been cut from the tubes they represent.

If any of the aforesaid tubes are made in long lengths and passed by the Surveyor in that condition, the number of tests required may be calculated on the number of tubes as made, notwithstanding that they may afterwards be cut up into shorter lengths.

### Table CIII.—Board of Trade Tensile Tests.

| Description of Material. | Minimum ultimate tensile strength, tons per square inch. | Maximum ultimate tensile strength, tons per square inch. | Elongation on 10 inches, per cent. |
|---|---|---|---|
| Plates not exposed to flame, . . . | 27 | 32 | 20% in 8 ins. |
| Plates that are exposed to flame, . | 26 | 30 | Not less than 23% for annealed plates. |
| Rivet bars, . . | 26 | 30 | Not less than 25%. |
| Stay bars, ordinary . | 27 | 32 | Not less than 20%. |
| Stay bars, combustion chamber . . | 26 | 32 | Not less than 23%. |
| Tube strips . . | 26 | 30 | About 25%.<br>Not less than 20%. |
| Rivets, . . . | 27 | 32 | Contraction of area about 60%. |
| Shell plates, special, on application up to 36 tons. | | | |

Steel tubes should be made of open-hearth acid steel, unless material of other quality has been specially approved for the purpose. Solid-drawn tubes of a thickness exceeding $\frac{1}{8}$ inch should be finished by the hot-drawn process, unless cold-drawing has been specially sanctioned, and all cold-drawn tubes should afterwards be efficiently annealed.

## STEEL BOILERS.

136. *Thickness of plates: drilling, welding, and annealing.*—The thickness of plates, other than tube strips, used in the construction of boilers should not be less than $\frac{5}{16}$ inch.

It is expected that the rivet holes will be drilled, and not punched. Plates that are drilled in place should be taken apart and the burr taken off, and the holes slightly countersunk from the outside.

Butt straps should be cut from plates, and not from bars.

Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed.

Local heating of the plates should be avoided, as many plates have failed from having been so treated.

All plates that have been flanged or locally heated, and all stays and stay tubes which have been locally heated, should be carefully annealed after being so treated.

137. *Cylindrical Boiler Shells.*—The Board of Trade consider that boilers well constructed, well designed, and made of good material should be allowed an advantage in the matter of working pressure over boilers inferior in any of the above respects, as, unless this is done, the superior boiler is placed at a disadvantage, and good workmanship and material will be discouraged. They have, therefore, caused the following rules to be prepared :—

When the cylindrical shells of boilers are made of material which has been duly tested and approved, with all the rivet holes drilled in place and all the seams fitted with double butt straps, each of at least five-eighths the thickness of the plates they cover, and all the seams at least double riveted with rivets having an allowance of not more than 87·5 per cent. over the single shear, provided that the boilers have been open to inspection during the whole period of construction, then 4·5 may be used as the factor of safety, the minimum actual tensile strength of the plates being used in calculating the working pressure.

When the above conditions are not complied with, the additions in the scale shown in footnote to Table CIV. should be made to the factor of safety, according to the circumstances of each case.

The joints referred to in the table as circumferential seams do not include the joints between the shell and end plates, which it is expected will be at least double riveted when the thickness of the shell exceeds $\frac{1}{2}$ inch.

When surveying boilers that have not been open to inspection during construction, the case should be submitted to the Board as to the factor to be used.

## RULES FOR STRENGTHS OF PARTS OF BOILERS.

**Board of Trade rules for cylindrical shells.**—In all calculations for strength of shells the *minimum* strength of plate, as disclosed by the tests, must be used.

To ascertain the strength of shell, the relative sectional areas of plate and rivet must first be determined by the following formulæ :—

**Rule 248.** $$\frac{(\text{Pitch} - \text{dia. of rivet}) \times 100}{\text{Pitch.}} = \left\{\begin{array}{l}\text{Percentage of strength}\\ \text{of plate at joint as com-}\\ \text{pared with solid plate.}\end{array}\right.$$

For maximum permissible pitch of rivets, &c., see formulæ and sketches.

**Rule 249.**

$$\left.\begin{array}{c}\dfrac{(\text{Area of rivet} \times \text{No. of rivets in pitch}) \times 100}{\text{Pitch} \times \text{thickness of plate}}\\ \text{or}\\ \dfrac{78{\cdot}54n \times d^2}{p \times t}\end{array}\right\} = \left\{\begin{array}{l}\text{Percentage of strength}\\ \text{of rivets as compared}\\ \text{with solid plate.}\end{array}\right.$$

If the rivets are in double shear, multiply the percentage thus found by 1·875.

In consequence of the low shearing strength of steel rivets the Board require that in all types of joints the *nominal* rivet section shall be not less than $\frac{27}{23}$ of the net plate section ; thus, in order that rivet section may be considered to have the same strength as plate section, their relation must be when the plate is 27-ton steel :—

In lap joints,—

**(Area of rivet × No. of rivets in pitch) × 23 = (Pitch − diameter of rivet) × thickness of plate × 27.***

And in butt joints,—

**(Area of rivet × No. of rivets in pitch × 1·875 × 23 = (Pitch − diameter of rivet) × thickness of plate × 27.***

or $$0{\cdot}7854d^2 \times n \times 1{\cdot}875 \times 23 = (p - d) \times 27t,$$

that is, $$1{\cdot}255nd^2 = (p - d)t.$$

---

* When plate of a higher tensile strength than 27 tons is used the actual strength in tons should be used in lieu of the value 27.

The working pressure per square inch that may be allowed on the safety valves is then given by,—

**Rule 250.** **Working Pressure** $=\dfrac{S\times\%\times(t-2)}{D\times 2{\cdot}8}$,

where $S$ = tensile strength of material in lbs. per square inch;
$\%$ = one of the two percentages, found by Rules 248 and 249, divided by 100;
$t$ = thickness of plate in 32nds of an inch;
$D$ = inside diameter of boiler in inches (inside diameter of outer strake, if any).

**Rules for riveting of longitudinal seams** when butted and fitted with double straps; the strength of the rivets against shear at 23 tons and the plate for tensile 27 tons:—

$n$ = the number of rivets per pitch of outer rows.
$x$ = the strength of joint as a fraction of the pitch.
$d$ = the diameter of rivet.
$p$ = pitch of outer rows.

**Rule 249a.** **Diameter of rivet** $=\dfrac{t}{1{\cdot}255n}\times\dfrac{x}{1-x}$.

**Rule 249b.** **Pitch of rivet** $=\dfrac{\text{diameter}}{1-x}$.

These are deduced from the following equations:—

(*a*) $$n\times{\cdot}7854d^2\times 1{\cdot}875\times 23=(p-d)27t,$$

that is, $$1{\cdot}255nd^2=(p-d)t.$$

(*b*) $$\frac{p-d}{p}=x.$$

*N.B.*—The Board of Trade require the rivets diameter in all cases to be not less than the thickness of plate.

**The B.M.E.D. & C. Committee's rule** for boiler shells is that adopted by the Board of Trade as above, but the factors are 2·67 when the longitudinal seams are made with double-butt straps and 2·87 when made with lap joints. The Committee approves the use of *basic* as well as acid steel, provided it is made by the *open-hearth* process and withstands the same mechanical tests.

**Butt Straps.**—The outer must have at least 62·5 per cent. of the strength of the shell plate, and the inner one should be $^4/_{32}$nds of an inch thicker than it.

When the number of rivets in the inner rows is double that in the outer, the thickness of outer strap

$$=\frac{5\times(p-d)}{8\times(p-2d)}\times \mathrm{T},$$

$d$ being the diameter of the rivets, T the thickness of shell plate.

When chain riveted, the distance between the outer and the next row of rivets should be not less than $0{\cdot}33p+0{\cdot}67d$ or $2d$, whichever is greater. When there are the full number of rivets in contiguous rows, the distance apart must be not less than $2d$. When zig-zag riveted, these distances are $0{\cdot}2p+1{\cdot}15d$ and $0{\cdot}2p+0{\cdot}4d$. In each case $p$ is the pitch in the outer rows.

The maximum pitch of rivets in longitudinal seams $=\mathrm{C}\times\mathrm{T}+1\frac{5}{8}$ inch. C is a coefficient as follows, and T is the thickness of plate or strap in inches.

| Number of Rivets per Pitch. | Coefficient C for Lap Joints. | Coefficient C for Butt Joints. |
|---|---|---|
| 1 | 1·31 | 1·75 |
| 2 | 2·62 | 3·50 |
| 3 | 3·47 | 4·63 |
| 4 | 4-14 | 5·52 |
| 5 | ... | 6·00 |

*N.B.*—The Board of Trade present limit of pitch is 12·25 ins.

**The British Corporation Rule for shell plates** is

$$\textbf{Working pressure}=\frac{(t-2)\times \mathrm{J}\times \mathrm{S}}{\mathrm{D}\times \mathrm{F}}.$$

In this case F = 2·75 when the butt straps are of equal width.
F = 2·85 ,, ,, ,, unequal ,,
F = 2·95 when the longitudinal seams are lapped.

**Lloyd's Register Old Rule** was as follows :—

$$\textbf{Working pressure}=\mathrm{C}\times(\mathrm{T}-2)\times \mathrm{B}\div \mathrm{D}.$$

In this case T is the thickness in sixteenths of an inch, and the values of C are 22, 21·25, and 20·5, depending on the nature of the joint as above.

This Society has adopted the rule recommended by the B.M.E.D. & C. Committee, the sizes derived from which differ but very slightly from those of the old rule.

The following Table shows the thickness of plates which will be required when steel of a higher tensile than 27 tons per square inch is to be used.

## Table CIV.—Relative Thickness of Boiler Shell Plates for Different Tensile Strengths.

| Minimum tensile test strength in tons per square inch. | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 | 36 | 37 | 38 |
| 0·50 | 0·483 | 0·466 | 0·450 | 0·486 | 0·422 | 0·409 | 0·397 | 0·386 | 0·375 | 0·365 | 0·355 |
| 0·60 | 0·579 | 0·559 | 0·540 | 0·524 | 0·507 | 0·491 | 0·476 | 0·463 | 0·450 | 0·438 | 0·427 |
| 0·70 | 0·675 | 0·652 | 0·630 | 0·611 | 0·591 | 0·573 | 0·551 | 0·540 | 0·525 | 0·511 | 0·498 |
| 0·80 | 0·772 | 0·745 | 0·720 | 0·699 | 0·675 | 0·655 | 0·635 | 0·617 | 0·600 | 0·584 | 0·569 |
| 0·90 | 0·868 | 0·838 | 0·810 | 0·784 | 0·760 | 0·736 | 0·714 | 0·694 | 0·675 | 0·657 | 0·640 |
| 1·00 | 0·965 | 0·931 | 0·900 | 0·873 | 0·844 | 0·818 | 0·793 | 0·771 | 0·750 | 0·730 | 0·711 |
| 1·10 | 1·061 | 1·024 | 0·990 | 0·961 | 0·929 | 0·900 | 0·872 | 0·848 | 0·825 | 0·803 | 0·782 |
| 1·20 | 1·157 | 1·117 | 1·080 | 1·048 | 1·013 | 0·982 | 0·925 | 0·925 | 0·900 | 0·876 | 0·853 |
| 1·30 | 1·254 | 1·211 | 1·170 | 1·135 | 1·097 | 1·064 | 1·031 | 1·003 | 0·975 | 0·949 | 0·924 |
| 1·40 | 1·350 | 1·304 | 1·260 | 1·223 | 1·182 | 1·145 | 1·110 | 1·080 | 1·050 | 1·022 | 0·996 |
| 1·50 | 1·447 | 1·397 | 1·350 | 1·310 | 1·266 | 1·227 | 1·190 | 1·157 | 1·125 | 1·095 | 1·067 |
| 1·60 | 1·543 | 1·490 | 1·440 | 1·397 | 1·351 | 1·309 | 1·269 | 1·234 | 1·200 | 1·168 | 1·138 |
| 1·70 | 1·640 | 1·583 | 1·530 | 1·484 | 1·435 | 1·391 | 1·348 | 1·311 | 1·275 | 1·241 | 1·209 |
| 1·80 | 1·736 | 1·676 | 1·620 | 1·572 | 1·519 | 1·473 | 1·428 | 1·388 | 1·350 | 1·314 | 1·280 |
| 1·90 | 1·833 | 1·769 | 1·710 | 1·659 | 1·604 | 1·554 | 1·507 | 1·465 | 1·425 | 1·387 | 1·350 |
| 2·00 | 1·929 | 1·862 | 1·800 | 1·746 | 1·688 | 1·636 | 1·586 | 1·542 | 1·500 | 1·460 | 1·421 |

139. *Openings in shells; doors, etc.*—The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally.

Compensating rings of at least the same effective sectional area as the plates cut out, and not less in thickness than the plates to which they are attached, should be fitted around all man-holes and openings, or the surrounding portion of the plates otherwise efficiently stiffened.

It is very desirable that the compensating rings around openings in flat surfaces be made of L- or T-bars. When a ring is not fitted around such an opening, and the plate is flanged for compensation, the total depth, D, of the flange should not be less than that given by the following equation :—

$$D = \sqrt{\text{width of opening} \times \text{thickness of plate}}.$$

Cast-iron doors should not be passed, and in all cases in which the surveyors find that cast-iron is employed in the construction of boilers in such a manner as to be subjected to the pressure of steam or water, they should report the circumstances to the Board of Trade.

The neutral part of the boiler shells under steam domes should be efficiently stiffened and stayed, as serious accidents have arisen from the want of such precautions.

In connection with openings in the ends of steam receivers, etc., the surveyors should bear in mind that, although the usual strengthening plate may be fitted around such openings, the end is still weaker than if no opening had been made, and that the fact of there being a plate or the flange of a neck-piece riveted around the opening may not remove the necessity for having the end suitably stayed.

## LLOYD'S RULES FOR BOILER SHELLS, &c.

### Quality and Testing of Boiler Steel.

**Section 4.**—1. When steel is used in the construction of boilers intended for vessels classed or proposed for classification in the Society's Register Book, the boilers shall be constructed in accordance with the requirements of the Rules, and the following conditions be fulfilled :—

2. **Process of Manufacture.**—Steel for marine boilers shall be made by the Open Hearth process, acid or basic.

3 **Freedom from Defects.**—The finished material shall be free from cracks, surface flaws, and lamination. It shall also have a workmanlike finish, and must not have been hammer-dressed.

4. **Testing and Inspection.**—The following tests and inspections shall be made at the place of manufacture prior to dispatch ; but, in the event of any of the material proving unsatisfactory in the course of being worked into boilers, such material shall be rejected, notwithstanding any previous certificate of satisfactory testing, and such further tests of the material from the same charge may be made as the Surveyor may consider desirable.

5. **Tensile Test-Pieces.**—The tensile strength and ductility shall be determined from Standard test-pieces cut lengthwise or crosswise from the rolled material. When material is annealed or otherwise treated before dispatch, the test-pieces shall be similarly and simultaneously treated with the material before testing.

*Plates, Angles, and Tee Bars.*—Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test-piece. The elongation shall be measured on a Standard test-piece having a gauge length of 8 inches.

For material more than ·875 inch in thickness, the width of the test·

piece between the gauge points shall not exceed 1½ inches; for material ·875 inch to ·375 inch in thickness, inclusive, the width shall not exceed 2 inches; for material less than ·375 inch in thickness, the width shall not be more than 2½ inches. In other respects the test-pieces shall conform generally to the Standard.

*Round Bars.*—Round bars may be tested either full size as rolled, or turned down when the diameter is considerable. The test-piece shall have a gauge length of not less than eight times its diameter, and a sectional area of not less than ¼ square inch. When enlarged ends are used, the length of the parallel portion shall not be less than nine times the reduced diameter.

Where bars are above one inch diameter, and are tested full size as rolled, or have been turned down, and the resulting test-piece is above 1 inch diameter, a gauge length of four times the diameter may be used if preferred by the manufacturer, in which case an increased elongation will be required, as specified in paragraph 7. When enlarged ends are used, the length of the parallel portion shall be not less than 4½ times the reduced diameter.

Any straightening of test-pieces which may be required shall be done cold.

**Mechanical Tests and Selection of Test-Pieces.**—Plates and bars for boilers shall comply with the following mechanical tests. All test pieces shall be selected by the Surveyor and tested in his presence, and he shall satisfy himself that the conditions herein described are fulfilled.

7. **Tensile Tests.** *Plates.*—The tensile breaking strength of steel plates for shells and girders, determined from Standard test-pieces, shall be between the limits of 28 and 32 tons per square inch. For plates intended for flanging or welding, and for combustion chambers and furnaces, the tensile breaking strength shall be between the limits of 26 and 30 tons per square inch. In the case of material for purposes in which tensile strength is not important, the tensile test may be dispensed with and the bend test only be made, if so specified by the builders and approved by the committee. The elongation, measured on a Standard test-piece having a gauge length of 8 inches, shall not be less than 20 per cent. for material of ·375 inch in thickness and upwards required to have a tensile breaking strength of 28 to 32 tons per square inch; and not less than 23 per cent. for material of ·375 inch in thickness and upwards required to have a tensile breaking strength of 26 to 30 tons per square inch.

*Stay, Angle, and Tee Bars.*—The tensile breaking strength of longitudinal stays and angle and tee bars shall be between the limits of 28 and 32 tons per square inch, with an elongation of not less than 20 per cent. of the gauge length measured on the Standard test-piece A or B, or 24 per cent. measured on the Standard test-piece C. For bars for combustion chamber stays the tensile breaking strength shall be between 26 and 30 tons per square inch, with an elongation of not less than 23 per cent. of the gauge length measured on the Standard test-piece B, or 28 per cent. measured on the Standard test-piece C.

For material under ·375 inch in thickness the elongation may be not more than 3 per cent. below the above-named elongation.

*Rivet Bars.*—The tensile breaking strength of rivet bars shall be between the limits of 26 and 30 tons per square inch of section, with an elongation of not less than 25 per cent. of the gauge length measured on the Standard test-piece B, or 30 per cent. measured on the Standard test-piece C. The bars may be tested the full size as rolled.

8. **Number of Tensile Tests.** *Plates.*—One tensile test shall be taken from each plate as rolled. For plates exceeding 2½ tons in weight one tensile test shall be taken from each end.

*Angle, Tee, Rivet, and Stay Bars*—At least two tensile tests for angle bars, tee bars, rivet bars, and stay bars shall be taken from each charge; but when the number of the bars, as rolled, from one charge exceeds 15, an additional tensile test shall be made for each further batch of 15 bars or portion thereof. In round bars of 1¾ inches diameter and under, the number shall be 50 in place of 15.

Should a tensile test-piece break outside the middle half of its gauge length, and the elongation be less than that required by the Rules, the test may, at the maker's option, be discarded and another test be made of the same plate or bar.

9. **Bend Tests.** *Cold Bends.*—Test-pieces shall be sheared lengthwise or crosswise from plates or bars, and shall not be less than 1½ inches wide, but for small bars the whole section may be used. For rivet bars bend tests are not required.

*Temper Bends.*—The test-pieces shall be similar to those used for cold bend tests. For temper bend tests the samples shall be heated to a blood red and quenched in water at a temperature not exceeding 80° F. The colour shall be judged indoors in the shade.

In all cold bend tests, and in temper bend tests on samples ·5 inch in thickness and above, the rough edge or arris caused by shearing may be removed by filing or grinding, and samples 1 inch in thickness and above may have the edges machined, but the test-pieces shall receive no other preparation. The test-pieces shall not be annealed unless the material from which they are cut is similarly annealed, in which case the test-pieces shall be similarly and simultaneously treated with the material before testing.

For both cold and temper bend the test-piece shall withstand, without fracture, being doubled over until the internal radius is equal to 1½ times the thickness of the test-piece, and the sides are parallel.

For small sectional material these bend tests may be made from the flattened bar.

Bend tests may be made either by pressure or by blows.

10. **Number of Bend Tests.** *Plates.*—One cold or temper bend test shall be taken from each plate as rolled. For plates exceeding 2½ tons in weight one bend test shall be taken from each end—one bend test to be temper and the other cold.

*Angle Bars.*—A cold or a temper bend test shall be made from each angle bar rolled.

*Stay Bars.*—A cold and temper bend test shall be made from every fifteen stay bars as rolled from each charge.

11. **Tests for Manufactured Rivets.**—Rivets selected by the Surveyor from the bulk shall withstand the following test:—

(*a*) The rivet shanks are to be bent cold, and hammered until the two parts of the shank touch without fracture on the outside of the bend.

(*b*) The rivet heads are to be flattened, while hot, without cracking at the edges. The heads are to be flattened until their diameter is 2½ times the diameter of the shank.

12. **Additional Tests before Rejection.**—Should any of the test-pieces first selected by the Surveyor not fulfil the test requirements, two further tests may be made from the same plate or bar, but should either of these fail, the plate or bar from which the test-pieces were cut shall be rejected. In all such cases further tests shall be made before any material from the same charge can be accepted.

13. **Branding.**—Every plate and bar shall be clearly and distinctly marked by the maker in two places with the Society's brand indicating that the material has complied with the Society's tests.

**Lloyd's Old Rules for cylindrical shells.**—(Section 12.)

**Rule 251. Working pressure (lbs. per sq. inch.)** $= \dfrac{C \times (T-2) \times B}{D}$

where D = mean diameter of shell in inches.

T = thickness of plate in *sixteenths of an inch.*

$$C^* = \begin{cases} 22 - & \text{when longitudinal seams have double butt straps of equal width.} \\ 21{\cdot}25 & \text{when longitudinal seams have double butt straps of unequal width, only covering one side outer row of nut.} \\ 20{\cdot}5 & \text{when longitudinal seams are lap joints.} \end{cases}$$

B = least percentage of strength of longitudinal joint, found as follows:—

For plate at joint, $B = \dfrac{p-d}{p} \times 100.$

For rivets at joint, $B = \dfrac{n \times a}{p \times t} \times 85$ (if rivets are of iron, $f = 70$),

where $p$ = pitch of rivets in inches;
$t$ = thickness of plate *in inches*;
$d$ = diameter of rivet holes in inches;
$n$ = number of rivets used per pitch in longitudinal joint;
$a$ = sectional area of rivet in square inches.

When rivets are in double shear $1{\cdot}75a$ is to be used in place of $a$.

Inside butt straps are to be at least three-quarters the strength of the longitudinal joint.

For the shells of superheaters or steam-chests enclosed in the uptakes, or exposed to the direct action of flame, the values of C should be only two-thirds of those given above.

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* *Note.*—If minimum tensile strength of plates is 28 tons; values of C may be correspondingly modified if strength is other than 28 tons.

Proper deductions are to be made for openings in shells; all manholes in shells to have compensating rings.

Shell plates under domes are to be adequately stayed.

## RULES FOR THICKNESS OF FLAT PLATES.

**Board of Trade** rules for flat surfaces.—The highest permissible pressures on flat surfaces are found by the following formula:—

Rule 252. Working pressure (lbs. per square inch) $= \dfrac{C \times (T+1)^{2}}{S-6}$*

where T = thickness of plate in *sixteenths of an inch*;
S = surface supported in square inches;
C = constant having values given in following table:—

Table CV.—**Board of Trade** Constants for Flat Surfaces.

| Description of attachment of stay. | Plates not exposed to heat or flame. | Plates exposed to heat or flame and in contact with steam. | Plates exposed to heat or flame and in contact with water. |
|---|---|---|---|
| Stays with nuts on both sides of plates and doubling strips, the latter of width equal to ⅔rds pitch and thickness equal to plates outside which they are riveted. | Up to ¾ in. thick, 240, and increasing to 264 for 1¼ thick. | The flat ends of all boilers, as far as the steam space extends, should be fitted with shield, or baffle plates where exposed to the hot gases in the uptake.<br>... | ... |
| Stays with nuts on both sides of plates and riveted washers,—latter having diameter equal to ⅔rds pitch and thickness equal to plates outside which they are riveted. | Up to ¾ in. thick, 210, and increasing to 231 for 1¼ thick. | ... | ... |
| Stays with nuts on both sides of plates and washers outside the plates,—latter having diameter three times that of stay and thickness ⅔rds that of plate they cover. | Up to ⅞ in. thick, 165, and increasing to 181 for 1⅜ thick. | 75 | ... |
| Stays fitted with nuts only on both sides of plates. | Up to ⅞ in. thick, 150, and increasing to 165 for 1½ thick. | 67·5 | ... |
| Stays screwed into plate and fitted with nuts. | ... | ... | 100 |
| Stays screwed into plate and riveted over or expanded. | 77 | 39·6 | Up to ½ in. thick, 66, and increasing to 70 for $\frac{11}{16}$ thick. |

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* *Note.*—Or, $T = \sqrt{\dfrac{P \times (S-6)}{C}} - 1$.

**The Rules of the B.M.E.D. & C. Committee for flat surfaces** are:—

**Rule 252a. Working pressure** $=(t-1)^2 \times C \div (a^2+b^2)$,

$t$ is the thickness in 32nds inch; $t_n$ that of washers and doublings; $a$ the distance apart of rows of stays whose pitch is $b$; C a coefficient.

C for screwed stays with ends riveted only, 50.
C ,, ,, ,, ,, nuts on outside of plates, 75, and when exposed to flame, 88.
C for stay tubes without nuts, 55; with nuts, 72.

When the stays pass through the plates and have nuts inside and outside nuts with washers of diameter 3·5 the diameter of stay and at least $0{\cdot}2 \times$ diameter of stay in thickness,

**Rule 252b. Working pressure** $=\frac{100}{a^2+b^2}\{(t-1)^2+0{\cdot}15t_n{}^2\}$.

When the washer is 0·66 the pitch and riveted to plate,

**Rule 252c. Working pressure** $=\frac{100}{a^2+b^2}\{(t-1)^2+0{\cdot}35t_n{}^2\}$.

When there are doubling straps in width 2/3, pitch and riveted to plate,

**Rule 252d. Working pressure** $=\frac{100}{a^2+b^2}\{(t-1)^2+0{\cdot}55t_n{}^2\}$.

**Front tube plates.**

**Rule 252e. Working pressure** $=C(t-1)^2 \div p^2$,

$p$ is the mean pitch and $C=38$ screwed only, and 49 with nuts.

**Wide spaces in front tube plate.**

**Rule 252f. Working pressure** $=C\{(t-1)^2+0{\cdot}55t_n{}^2\} \div a^2+b^2$,

$a$ is the horizontal pitch of stay tubes across the wide space.
$b$ is the vertical pitch in the bounding rows.
$C=55$ when screwed only, and 72 with nuts; when there are nuts only on alternate rows, $C=63$.

**Compression on tube plates.**

**Rule 252g. Working pressure** $=\frac{875 \times (D-d) \times t}{B \times D}$,

where D is the horizontal distance apart of the tubes.
$d$ is the internal diameter of tubes.
B the width of combustion chamber.

## Table CVI.—Pitch of Stays and Area of Flat Surfaces of Combustion Chambers (B.M.E.D. & C.C.)

| Pressure in lbs. per square inch. | Stays screwed through plates and fitted with nuts at ends. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Plates $^{16}/_{32}$ inch. | | Plates $^{18}/_{32}$ inch. | | Plates $^{20}/_{32}$ inch. | | Plates $^{22}/_{32}$ inch. | | Plates $^{24}/_{32}$ inch. | |
| | Pitch. | Surf. | Pitch. | Surf. | Pitch. | Surf. | Pitch. | Surf. | Pitch. | Surf. |
| 150 | 7·49 | 56·1 | 8·50 | 72·2 | 9·50 | 90·2 | 10·49 | 110 | 11·49 | 132 |
| 155 | 7·37 | 54·3 | 8·37 | 70·0 | 9·34 | 87·3 | 10·30 | 106 | 11·31 | 128 |
| 160 | 7·25 | 52·6 | 8·22 | 67·6 | 9·20 | 84·6 | 10·15 | 103 | 11·09 | 123 |
| 165 | 7·14 | 51·0 | 8·11 | 65·7 | 9·06 | 82·0 | 10·00 | 100 | 10·95 | 120 |
| 170 | 7·04 | 49·5 | 7·98 | 63·7 | 8·92 | 79·6 | 9·86 | 97·3 | 10·73 | 116 |
| 175 | 6·94 | 48·1 | 7·87 | 61·9 | 8·79 | 77·3 | 9·72 | 94·5 | 10·63 | 113 |
| 180 | 6·84 | 46·8 | 7·76 | 60·2 | 8·67 | 75·1 | 9·57 | 91·6 | 10·48 | 110 |
| 185 | 6·75 | 45·5 | 7·65 | 58·5 | 8·56 | 73·2 | 9·46 | 89·4 | 10·34 | 107 |
| 190 | 6·66 | 44·3 | 7·55 | 57·0 | 8·44 | 71·2 | 9·33 | 87·0 | 10·20 | 104 |
| 195 | 6·57 | 43·2 | 7·45 | 55·5 | 8·33 | 69·4 | 9·21 | 84·9 | 10·05 | 101 |
| 200 | 6·49 | 42·1 | 7·36 | 54·2 | 8·23 | 67·7 | 9·09 | 82·7 | 9·95 | 99·1 |
| 205 | 6·41 | 41·1 | 7·27 | 52·8 | 8·13 | 66·0 | 8·98 | 80·6 | 9·83 | 96·6 |
| 210 | 6·39 | 40·9 | 7·19 | 51·6 | 8·03 | 64·4 | 8·87 | 78·7 | 9·72 | 94·4 |
| 215 | 6·33 | 40·0 | 7·10 | 50·4 | 7·93 | 62·9 | 8·77 | 76·9 | 9·60 | 92·2 |
| 220 | ... | ... | 7·01 | 49·1 | 7·84 | 61·5 | 8·67 | 75·2 | 9·49 | 90·1 |
| 225 | ... | ... | 6·94 | 48·1 | 7·76 | 60·2 | 8·57 | 73·5 | 9·39 | 88·1 |
| 230 | ... | ... | ... | ... | 7·67 | 58·8 | 8·48 | 71·9 | 9·28 | 86·2 |
| 235 | ... | ... | ... | ... | 7·59 | 57·6 | 8·39 | 70·3 | 9·19 | 84·4 |
| 240 | ... | ... | ... | .. | ... | ... | 8·30 | 68·9 | 9·09 | 82·6 |
| 245 | ... | ... | ... | ... | ... | ... | 8·20 | 67·3 | 9·00 | 81·0 |
| 250 | ... | ... | ... | ... | ... | ... | 8·13 | 66·1 | 8·91 | 79·3 |

*N.B.*—When plates are not exposed to flame, multiply the *surfaces* above by 1·173, and the pitches by 1·083.

## Table CVII.—Pitch of Stays supporting Flat Plates not exposed to Flame by B.M.E.D. & C.C. Rule. (Thicknesses in 32nds of an inch.)

| Working Pressure, lbs. | Stays are fitted with nuts inside and out. Washer under outer is 3·5 diameter of stay. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Plate 20. Washer 14. | Plate 22. Washer 15. | Plate 24. Washer 16. | Plate 26. Washer 17. | Plate 28. Washer 19. | Plate 30. Washer 20. | Plate 32. Washer 22. | Plate 34. Washer 22. | Plate 36. Washer 22. | Plate 38. Washer 22. | Plate 40. Washer 22. |
| 150 | 11·40 | 12·58 | 13·75 | 14·92 | 16·16 | 17·42 | 18·46 | 19·73 | 20·80 | ... | ... |
| 155 | 11·20 | 12·38 | 13·52 | 14 68 | 15·88 | 17·08 | 18·22 | 19·41 | 20·47 | ... | ... |
| 160 | 11·04 | 12·18 | 13·31 | 14·45 | 15·64 | 16·77 | 17·98 | 19·11 | 20·14 | 21·23 | ... |
| 165 | 10·04 | 12·00 | 13·11 | 14·23 | 15·33 | 16·52 | 17·70 | 18·82 | 19·90 | 20·92 | ... |
| 170 | 10·71 | 11·82 | 12·91 | 14·02 | 15·17 | 16·28 | 17·44 | 18·54 | 19·54 | 20·61 | 21·65 |
| 175 | 10·56 | 11·65 | 12·73 | 14·81 | 14 95 | 16·23 | 17·19 | 18·27 | 19·26 | 20·31 | 21·34 |
| 180 | 10·41 | 11·49 | 12·58 | 13·62 | 14·75 | 15·81 | 16·95 | 18·01 | 18·98 | 20·02 | 21·04 |
| 185 | 10·27 | 11·33 | 12·40 | 13·45 | 14·55 | 15·60 | 16·72 | 17·76 | 18·73 | 19·73 | 20·78 |
| 190 | 10·13 | 11·18 | 12·21 | 13·29 | 14·37 | 15·40 | 16 50 | 17·53 | 18·49 | 19·44 | 20·48 |
| 195 | 10·00 | 11·04 | 12·05 | 13·15 | 14·18 | 15·20 | 16·28 | 17·31 | 18·25 | 19·21 | 20·22 |
| 200 | 9·87 | 10·90 | 11·91 | 12·92 | 14·00 | 15·01 | 16·08 | 17·09 | 18·02 | 18·99 | 19·96 |
| 205 | 9·75 | 10·77 | 11·76 | 12·76 | 13·82 | 14·83 | 15·88 | 16 83 | 17·80 | 18·76 | 19·71 |
| 210 | 9·63 | 10·64 | 11·62 | 12 61 | 13·65 | 14·65 | 15·69 | 16·68 | 17·58 | 18·53 | 19·46 |
| 215 | ... | 10 51 | 11·48 | 12·46 | 13·49 | 14·47 | 15·51 | 16·49 | 17·37 | 18·32 | 19·25 |
| 220 | ... | ... | 11·35 | 12·32 | 13·34 | 14·30 | 15·33 | 16·30 | 17·17 | 18·11 | 19·04 |
| 225 | ... | ... | ... | 12·18 | 13·19 | 14·15 | 15·16 | 16·12 | 16·99 | 17·91 | 18·83 |
| 230 | ... | ... | ... | 12·05 | 13·05 | 14·00 | 14·99 | 15·94 | 16·81 | 17·72 | 18·62 |
| 235 | ... | ... | ... | ... | 12·91 | 13·85 | 14·83 | 15·77 | 16·63 | 17·52 | 18·42 |
| 240 | ... | ... | ... | ... | 12·77 | 13·71 | 14·68 | 15·60 | 16·45 | 17·33 | 18·22 |
| 245 | ... | ... | ... | ... | ... | 13·57 | 14·53 | 15·44 | 16·28 | 17·16 | 18·04 |
| 250 | ... | ... | ... | ... | ... | 13·43 | 14·39 | 15·28 | 16·12 | 17·00 | 17·86 |

*N.B.*—For surface pitch$^2 = \dfrac{a^2 + b^2}{2}$.

## Table CVIII.—Pitch of Stays supporting Flat Plates when fitted with Washers 0·66 of the Pitch in Diameter Riveted to them outside. By B.M.E.D. & C.C. Rule.

| Working Pressure, lbs. | Thicknesses of plates and washers in 32nds of an inch. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Plate 20. | | Plate 22. | | Plate 24. | | Plate 26. | | Plate 28. | | Plate 30. | | Plate 32. | |
| | Washer | | Washer | | Washer | | Washer | | Washer | | Washer | | Washer | |
| | 14. | 20. | 16. | 22. | 18. | 24. | 20. | 26. | 21. | 28. | 22. | 30. | 23. | 32. |
| 150 | 11·97 | 12·92 | 13·30 | 14·26 | 14·63 | 15·60 | 15·98 | 16·95 | 17·16 | 18·26 | 18·35 | 19·63 | 19·56 | 20·98 |
| 160 | 11·59 | 12·51 | 12·88 | 13·80 | 14·16 | 15·10 | 15·46 | 16·41 | 16·61 | 17·67 | 17·76 | 19·01 | 18·94 | 20·31 |
| 170 | 11·24 | 12·14 | 12·50 | 13·41 | 13·75 | 14·65 | 15·00 | 15·93 | 16·12 | 17·15 | 17·24 | 18·44 | 18·38 | 19·73 |
| 180 | 10·95 | 11·80 | 12·15 | 13·02 | 13·36 | 14·24 | 14·59 | 15·47 | 15·66 | 16·68 | 16·75 | 17·92 | 17·86 | 19·15 |
| 190 | 10·64 | 11·48 | 11·83 | 12·66 | 13·00 | 13·86 | 14·19 | 15·06 | 15·24 | 16·24 | 16·30 | 17·44 | 17·38 | 18·64 |
| 200 | 10·37 | 11·19 | 11·24 | 12·35 | 12·67 | 13·51 | 13·83 | 14·68 | 14·86 | 15 82 | 15·89 | 17·00 | 16·94 | 18·17 |
| 210 | 10·11 | 10·92 | 10·99 | 12·05 | 12·33 | 13·19 | 13·50 | 14·33 | 14·50 | 15·44 | 15·51 | 16·59 | 16·53 | 17·73 |
| 220 | ... | 10·68 | 10·73 | 11·78 | 12·08 | 12·89 | 13·31 | 14·11 | 14·17 | 15·08 | 15·15 | 16·21 | 16·14 | 17·33 |
| 230 | ... | ... | 10·52 | 11·53 | 11·81 | 12·60 | 12·90 | 13·69 | 13·86 | 14·74 | 14·83 | 15·85 | 15·81 | 16·95 |
| 240 | ... | ... | ... | 11 26 | 11·57 | 12·33 | 12·62 | 13·41 | 13·56 | 14·43 | 14·50 | 15·52 | 15·47 | 16·59 |
| 250 | ... | ... | ... | ... | 11·33 | 12·08 | 12·37 | 13·13 | 13·29 | 14·15 | 14·21 | 15·20 | 15·15 | 16·25 |
| 260 | ... | ... | ... | ... | ... | 11·85 | 12·13 | 12·86 | 13·03 | 13·89 | 13·93 | 14·91 | 14·86 | 15·93 |
| 270 | ... | ... | ... | ... | ... | ... | 11·90 | 12·63 | 12·78 | 13 61 | 13·67 | 14 63 | 14·58 | 15 63 |
| 280 | ... | ... | ... | ... | ... | ... | ... | 12·41 | 12·54 | 13·36 | 13·43 | 14·37 | 14·32 | 15·31 |
| 290 | ... | ... | ... | ... | ... | ... | ... | ... | 12·33 | 13·13 | 13·20 | 14·13 | 14·08 | 15·09 |
| 300 | ... | ... | ... | ... | ... | ... | ... | ... | ... | 12·91 | 12·97 | 13·89 | 13·83 | 14·84 |

*N.B.*—For surface $=\text{pitch}^2=\dfrac{a^2+b^2}{2}$.

## Table CIX.—Boiler Shells of 28 tons Tensile Steel and 80 per cent. Longitudinal Jointing: Working Pressure in lbs. per sq. inch allowed by B.M.E.D. & C. Committee.

| Diameter of the Shell. | Thickness of Shell Plates in 32nds of an inch. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16. | 17. | 18. | 19. | 20. | 21. | 22. | 23. | 24. | 25. | 26. | 27. | 28. | 29. |
| Feet. | | | | | | | | | | | | | | |
| 6·00 | 163 | 174 | 186 | 198 | 209 | 221 | 233 | 244 | 256 | .. | .. | .. | .. | .. |
| 6·25 | 156 | 168 | 179 | 190 | 201 | 212 | 224 | 235 | 246 | 257 | .. | .. | .. | .. |
| 6·50 | 150 | 161 | 172 | 183 | 193 | 204 | 215 | 226 | 236 | 247 | 258 | .. | .. | .. |
| 6·75 | 145 | 155 | 165 | 176 | 186 | 196 | 207 | 217 | 228 | 238 | 248 | 259 | .. | .. |
| 7·00 | 140 | 150 | 160 | 170 | 180 | 190 | 200 | 210 | 220 | 230 | 240 | 250 | 260 | .. |
| 7·25 | 135 | 144 | 154 | 163 | 173 | 183 | 192 | 202 | 212 | 221 | 231 | 241 | 250 | 260 |
| 7·50 | .. | 140 | 149 | 158 | 168 | 177 | 186 | 196 | 205 | 214 | 224 | 233 | 242 | 252 |
| 7·75 | .. | 135 | 144 | 153 | 162 | 171 | 180 | 189 | 198 | 207 | 216 | 225 | 234 | 244 |
| 8·00 | .. | .. | 140 | 148 | 157 | 166 | 174 | 183 | 192 | 201 | 209 | 218 | 227 | 236 |
| 8·25 | .. | .. | 135 | 144 | 152 | 161 | 169 | 178 | 186 | 195 | 203 | 212 | 220 | 229 |
| 8·50 | .. | .. | .. | 139 | 148 | 156 | 164 | 172 | 181 | 189 | 197 | 205 | 214 | 222 |
| 8·75 | .. | .. | .. | 136 | 144 | 152 | 160 | 168 | 176 | 184 | 192 | 200 | 208 | 216 |
| 9·00 | .. | .. | .. | .. | 139 | 147 | 155 | 163 | 171 | 178 | 186 | 194 | 202 | 209 |
| 9·25 | .. | .. | .. | .. | 136 | 143 | 151 | 158 | 166 | 173 | 181 | 189 | 196 | 204 |
| 9·50 | .. | .. | .. | .. | .. | 140 | 147 | 154 | 162 | 169 | 176 | 184 | 191 | 198 |
| 9·75 | .. | .. | .. | .. | .. | 136 | 143 | 150 | 157 | 165 | 172 | 179 | 186 | 193 |
| 10·00 | .. | .. | .. | .. | .. | .. | 140 | 147 | 154 | 161 | 168 | 175 | 182 | 189 |
| 10·25 | .. | .. | .. | .. | .. | .. | 136 | 143 | 150 | 156 | 163 | 170 | 177 | 184 |
| 10·50 | .. | .. | .. | .. | .. | .. | .. | 139 | 146 | 153 | 160 | 166 | 173 | 180 |

*N.B.*—The pressure allowed by the Register Societies and Board of Trade is 97 per cent. of the above.

## Table CIXa.—Boiler Shells of 32 tons Tensile Steel and 84 per cent. Longitudinal Joints: Working Pressure in lbs. per sq. inch allowed by the B.M.E.D. & C. Committee.

| Diameter of the Shell. | Thickness of Shell Plates in 32nds of an inch. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 22. | 24. | 26. | 28. | 30. | 32. | 34. | 36. | 38. | 40. | 42. | 44. | 46. | 48. |
| Feet. | | | | | | | | | | | | | | |
| 8·0 | 209 | 230 | 251 | 272 | 293 | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| 8·5 | 197 | 217 | 237 | 256 | 276 | 296 | .. | .. | .. | .. | .. | .. | .. | .. |
| 9·0 | 186 | 205 | 223 | 242 | 260 | 279 | 298 | .. | .. | .. | .. | .. | .. | .. |
| 9·5 | 176 | 194 | 212 | 229 | 247 | 265 | 282 | 300 | .. | .. | .. | .. | .. | .. |
| 10·0 | 168 | 184 | 201 | 218 | 235 | 252 | 268 | 285 | 302 | .. | .. | .. | .. | .. |
| 10·5 | 160 | 176 | 192 | 208 | 224 | 240 | 256 | 272 | 288 | 304 | .. | .. | .. | .. |
| 11·0 | 152 | 167 | 183 | 198 | 213 | 229 | 244 | 259 | 274 | 290 | .. | .. | .. | .. |
| 11·5 | 146 | 160 | 175 | 189 | 204 | 219 | 233 | 248 | 262 | 277 | 292 | .. | .. | .. |
| 12·0 | .. | 154 | 168 | 182 | 196 | 210 | 224 | 238 | 252 | 266 | 280 | 294 | .. | .. |
| 12·5 | .. | 147 | 161 | 174 | 187 | 201 | 214 | 228 | 241 | 255 | 268 | 281 | 295 | .. |
| 13·0 | .. | .. | 155 | 167 | 180 | 193 | 206 | 219 | 232 | 245 | 258 | 271 | 283 | 296 |
| 13·5 | .. | .. | 149 | 161 | 174 | 186 | 199 | 211 | 223 | 236 | 248 | 261 | 273 | 286 |
| 14·0 | .. | .. | 144 | 156 | 168 | 180 | 192 | 204 | 216 | 228 | 240 | 252 | 264 | 276 |
| 14·5 | .. | .. | .. | 151 | 162 | 174 | 185 | 197 | 209 | 220 | 232 | 243 | 255 | 267 |
| 15·0 | .. | .. | .. | 145 | 156 | 168 | 179 | 190 | 201 | 213 | 224 | 235 | 246 | 257 |
| 15·5 | .. | .. | .. | .. | 151 | 162 | 173 | 184 | 195 | 206 | 216 | 227 | 238 | 249 |
| 16·0 | .. | .. | .. | .. | 147 | 157 | 168 | 178 | 189 | 199 | 210 | 220 | 231 | 241 |
| 16·5 | .. | .. | .. | .. | 142 | 152 | 163 | 173 | 183 | 193 | 203 | 214 | 224 | 234 |
| 17·0 | .. | .. | .. | .. | .. | 148 | 158 | 168 | 178 | 188 | 197 | 207 | 217 | 227 |
| 17·5 | .. | .. | .. | .. | .. | 144 | 153 | 163 | 173 | 182 | 192 | 202 | 211 | 221 |
| 18·0 | .. | .. | .. | .. | .. | .. | 149 | 158 | 168 | 177 | 186 | 196 | 205 | 214 |
| 18·5 | .. | .. | .. | .. | .. | .. | 145 | 154 | 163 | 172 | 181 | 191 | 200 | 209 |

*N.B.*—The pressure allowed by the Register Societies and Board of Trade is 97 per cent. of the above.

140. Hemispherical ends subject to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness.

141. Ends of steam receivers which are dished and flanged in a hydraulic press, the whole end being operated upon at each heat, may be passed without stays, provided that the radius of the end is not greater than the diameter of the shell to which it is attached and does not exceed 4 feet, that the outer radius of the flange at the root is not less than 3 inches, that the end is made of mild steel of the usual quality, and that it is efficiently annealed after the completion of the setting and flanging ; but the maximum pressure allowed without stays should not exceed that found by the following formula :—

$$\frac{90{,}000 \times T \times h}{D^2} = \text{working pressure} ;$$

where T = thickness of end, in inches ;

D = diameter of shell, inside, in inches ;

$h$ = distance, in inches, from the centre of the end on the inner side to a straight line passing through the points where the curve of the end due to the inner radius, and produced, would intersect the sides of the cylindrical shell, produced if necessary, at the diameter D, and

$$= R - \sqrt{R^2 - \frac{D^2}{4}}, \text{ in which } R = \text{inner radius of end, in inches.}$$

If there is a man-hole in the end, T should exceed that found by the formula by $\frac{1}{8}$ inch.

The above instructions do not apply to dished ends of vertical donkey boilers, which are subject to the thrust of the uptake in addition to the pressure of steam.

142. Stays, properly distributed, should be fitted to dished ends which are not of the thickness required for flat ends, or which do not comply with the requirements stated in Section 141 ; but, if the ends are sufficient for the pressure needed, when considered as portions of spheres, the stays, if of solid steel, may have a nominal stress of 18,000 lbs. per square inch of net section. If dished ends are not equal to the pressure needed when considered as portions of spheres, they should be stayed as flat surfaces.

**The B.M.E.D. & C. Committee's rule for dished ends is :—**

**Rule 252h.** **Working pressure** $= 15 \times S(t-1) \div R$,

$t$ is the thickness in *thirty-seconds of an inch*, R the radius of curvature in inches, S the minimum tensile strength of plates in tons.

The inside curvature of flange must have a radius not less than $4 \times t$, and in no case less than 2·5 inches.

## LLOYD'S OLD RULES FOR FLAT SURFACES.

### Table CX.—**Lloyd's Constants for Flat Surfaces.**

| Description of attachment of stay. | Value of C. |
|---|---|
| Screw stays with riveted head,— | |
| Plates $\frac{7}{16}$-inch thick and under, . . . . | 90 |
| Plates over $\frac{7}{16}$-inch thick, . . . . | 100 |
| Screw stays fitted with nuts,— | |
| Plates $\frac{7}{16}$-inch thick and under, . . . . | 110 |
| Plates over $\frac{7}{16}$-inch and under $\frac{9}{16}$-inch, . | 120 |
| Plates $\frac{9}{16}$-inch and over, . . . . | 135 |
| Stays fitted with double nuts, . . . . | 175 |
| Stays fitted with double nuts and with washers of diameter equal to $\frac{1}{3}$rd the pitch, and of thickness equal to half that of the plate they cover, . . | 185 |
| Stays fitted with double nuts and with outside riveted washers of diameter equal to $^2/_5$ths pitch, and of thickness equal to half that of the plate they cover, | 200 |
| Stays fitted with double nuts and with outside riveted washers of diameter equal to $\frac{2}{3}$rds pitch, and of thickness equal to that of the plate they cover, . | 220 |
| Stays with double nuts and outside doubling strips of width equal to $\frac{2}{3}$rds distance between rows of stays, and thickness equal to that of the plate they cover, when P is taken as the pitch of rows, . | 220 |
| Stays with double nuts and outside doubling strips of width equal to $\frac{2}{3}$rds distance between rows of stays, and thickness equal to that of the plate they cover, when P is taken as the pitch of stays in the rows, . . . . . . . | 240 |

*N.B.*—Lloyd's Register has adopted the rules of the B.M.E.D. & C. Committee.

## Table CXI.—Corrugated Furnaces (Fox, Morison, Deighton, Purves): Working Pressure in lbs. per sq. inch allowed by B.M.E.D. & C. Committee.

| Diameter at Bottom of Corrugations. | Thicknesses in 32nds of an inch. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 12. | 13. | 14. | 15. | 16. | 17. | 18. | 19. | 20. | 21. | 22. | 23. | 24. | 25. |
| ins. | | | | | | | | | | | | | | |
| 30 | 176 | 192 | 208 | 224 | 240 | 256 | 272 | 288 | 304 | ... | ... | ... | ... | ... |
| 31 | 170 | 185 | 201 | 216 | 232 | 247 | 263 | 278 | 294 | 309 | ... | ... | ... | ... |
| 32 | 165 | 180 | 195 | 210 | 225 | 240 | 255 | 270 | 285 | 300 | ... | ... | ... | ... |
| 33 | 160 | 174 | 189 | 203 | 218 | 232 | 247 | 261 | 276 | 291 | 305 | ... | ... | ... |
| 34 | 155 | 169 | 183 | 197 | 211 | 226 | 240 | 254 | 268 | 282 | 296 | 310 | ... | ... |
| 35 | 150 | 164 | 178 | 192 | 205 | 219 | 233 | 246 | 260 | 274 | 288 | 301 | ... | ... |
| 36 | 146 | 160 | 173 | 186 | 200 | 213 | 226 | 240 | 253 | 266 | 280 | 293 | 306 | ... |
| 37 | 143 | 156 | 169 | 182 | 195 | 208 | 221 | 234 | 247 | 260 | 273 | 286 | 299 | ... |
| 38 | ... | 151 | 164 | 176 | 189 | 202 | 214 | 227 | 240 | 252 | 265 | 277 | 290 | 303 |
| 39 | ... | 147 | 160 | 172 | 184 | 196 | 209 | 221 | 233 | 246 | 258 | 270 | 283 | 295 |
| 40 | ... | 144 | 156 | 168 | 180 | 192 | 204 | 216 | 228 | 240 | 252 | 264 | 276 | 288 |
| 41 | ... | ... | 152 | 163 | 175 | 187 | 199 | 210 | 222 | 234 | 245 | 257 | 269 | 281 |
| 42 | ... | ... | 148 | 160 | 171 | 183 | 194 | 205 | 217 | 228 | 240 | 251 | 263 | 274 |
| 43 | ... | ... | 145 | 156 | 167 | 178 | 189 | 200 | 212 | 223 | 234 | 245 | 256 | 267 |
| 44 | ... | ... | ... | 152 | 163 | 174 | 185 | 196 | 207 | 218 | 229 | 240 | 251 | 261 |
| 45 | ... | ... | ... | 149 | 160 | 170 | 181 | 192 | 202 | 213 | 224 | 234 | 245 | 256 |
| 46 | ... | ... | ... | 146 | 156 | 166 | 177 | 187 | 198 | 208 | 219 | 229 | 240 | 250 |
| 47 | ... | ... | ... | ... | 153 | 163 | 174 | 183 | 194 | 204 | 214 | 224 | 235 | 245 |
| 48 | ... | ... | ... | ... | 150 | 160 | 170 | 180 | 190 | 200 | 210 | 220 | 230 | 240 |
| 49 | ... | ... | ... | ... | 147 | 156 | 166 | 176 | 186 | 196 | 206 | 215 | 224 | 234 |
| 50 | ... | ... | ... | ... | 144 | 153 | 163 | 172 | 182 | 192 | 201 | 210 | 219 | 229 |
| 51 | ... | ... | ... | ... | ... | 150 | 160 | 169 | 178 | 188 | 197 | 207 | 216 | 225 |
| 52 | ... | ... | ... | ... | ... | 147 | 157 | 166 | 175 | 184 | 193 | 203 | 212 | 221 |

*N.B.*—For the working pressure on a bulb suspension furnace multiply by 1·065.

**Lloyd's old rules for flat surfaces.**—The strength of flat plates supported by stays is given by the following formula :—

**Rule 253.**

$$\text{Working pressure (lbs. per square inch)} = \frac{C \times T^2}{P^2},$$

where T = thickness of plate in *sixteenths of an inch* ;
C = constant having values given in preceding Table.
$P^2$ = square of pitch in inches. If pitches in rows and between rows are unequal, the mean of the two squares to be taken.

In the case of front plates of boilers in the steam space the values of C must be reduced 20 per cent. if the plates are not shielded from the direct action of heat.

If flat plates are strengthened by doubling plates securely riveted to them, and having a thickness of not less than two-thirds of the plates, the following formula may be used :—

**Rule 254.**

$$\text{Working pressure (lbs. per square inch)} = \frac{C \times (T + 0{\cdot}5t)^2}{P^2},$$

where $t$ = thickness of doubling plate in *sixteenths of an inch.*

For tube-plates in the nest of tubes, C must be taken in Rule 253 as 140, and P as the *mean* pitch of stay tubes.

For the wide spaces between nests of tubes, P is to be taken as the horizontal distance from centre to centre of the bounding rows of tubes and C, as follows :—

| Pitch of Stay Tubes in bounding rows. | When tubes have no nuts outside plates. | When tubes are fitted with nuts outside plates. |
|---|---|---|
| Where there are two plain tubes between each stay tube, . . . | 120 | 130 |
| Where there is one plain tube between each stay tube, . . . | 140 | 150 |
| Where every tube in these rows is a stay tube, and nuts when fitted are on alternate tubes, . . . | 160 | 170 |

## RULES FOR PLATES IN COMPRESSION.

**Board of Trade rule regarding compressive stresses on tube-plates.**

**Rule 255.**

$$\textbf{Working pressure (lbs. per square inch)} = \frac{(D - d)\ T \times 28{,}000}{W \times D},$$

where $D$ = least horizontal distance between centres of tubes, in inches ,
$d$ = inside diameter of ordinary tube, in inches ;
$T$ = thickness of tube-plate, *in inches* ;
$W$ = inside width of combustion chamber, in inches,—from tube-plate to back of box, or from tube-plate to tube-plate in double-ended boilers, with common combustion chambers.

**Lloyd's rule regarding compressive stresses on tube-plates.**

**Rule 256.**

$$T = \frac{P \times W \times D}{1750 \times (D - d)},$$

**Rule 256a.** **Rule of the B.M.E.D. & C. Committee.**

$$\textbf{Working pressure} = \frac{875 \times (D - d) \times t}{W \times D},$$

$t$ is the thickness in *thirty-seconds of an inch.*

## RULES FOR GIRDERS.

**Board of Trade rule for girders** supporting combustion chamber, tops, &c.

**Rule 257.**

$$\textbf{Working pressure (lbs. per square inch)} = \frac{C \times d^2 \times T}{(W - P)\ D \times L},$$

where $W$ = width of combustion chamber, in inches ;
$P$ = pitch of supporting bolts, in inches ;
$D$ = distance from centre to centre of girders, in inches ;
$L$ = length of girder, in feet ;
$d$ = depth of girder, in inches ;
$T$ = thickness of girder ,,
$N$ = number of supporting bolts ;

$C = \dfrac{N + 1320}{N + 1}$ when number of bolts is odd ;

$C = \dfrac{(N \times 1)1320}{N + 2}$ ,, ,, even.

**B.M.E.D. & C. Committee rule for girders** supporting combustion chamber, tops, &c.

**Rule 258.**

$$\text{Working pressure} = \frac{C \times d^2 \times T}{(L-P)\,D \times L} \times \frac{S}{28},$$

where L = width between tube-plates or tube-plate and back plate of chamber;
P = pitch of stays in girder;
D = distance from centre to centre of girder;
$d$ = depth of girder at centre;
} All in inches.
T = thickness of girder at centre in *thirty-seconds of an inch.*
S is the minimum tensile strength in tons of the plates forming the girder. When it is a forging, S is 24 for iron and 28 for steel.

$$C = \frac{n}{n+1} \times 495 \text{ when the number of stays to each girder is odd;}$$

$$C = \frac{n+1}{n+2} \times 495 \text{ when it is even; } n \text{ being the number.}$$

## RULES FOR STAYS.

**Admiralty requirements as regards stays.**—At the test pressure steel stays under 1½ inches diameter may carry a load of 16,000 lbs. per square inch of net section, and those of 1½ inches diameter and upwards, 18,000 lbs. Rivets or bolts used for securing stays must be at least 25 per cent. stronger than the stays.

**Board of Trade rules relating to stays.**—Steel stays that have been welded must not be used.

**Rule 259.** Solid steel stays may be allowed a working stress of 9000 lbs. per square inch of net section.

**Welded** steel stays may not be used, but this does not apply to stay tubes.

When the threads of longitudinal stays are finer than six per inch, the depth of the external nuts should be at least 1¼ times the diameter of the stay.

144. Stay tubes made of steel which has been tested may be allowed a stress not exceeding 7500 lbs. per square inch of net section, provided that their net thickness is in no case less than ¼ inch.

154. In the case of new boilers, the Surveyor may allow a stress not exceeding 7000 lbs. per square inch of net section on solid iron screwed stays supporting flat surfaces.

Iron screwed stays in the combustion chambers resist the cross-breaking stresses, to which they are subjected, better than steel; a stress of 9000 lbs. per square inch may be allowed on such stays, provided the bars have been tested and have a tensile strength of not less than $21\frac{1}{2}$ tons and an elongation, in 8 inches, of not less than 27 per cent.

155. A stress of 6000 lbs. may be allowed on the net section of iron stay tubes, provided that the net thickness is in no case less than $\frac{1}{4}$ inch.

156. The thickness of ordinary smoke tubes made of iron should not be less than that found by the following formula :—

$$T = \frac{\text{W.P.} \times D}{9000} + {\cdot}085,$$

where T = thickness of tube, in inches ;
D = outside diameter of tube, in inches.

**B.M.E.D. & C. Committee rules for stays** are as follows :—

(i.) Screw stays, 9 threads per inch of steel or tested iron—

**Rule 259a.**

$$\textbf{Working pressure} = (d - 0{\cdot}267)^2 \times 8250 \div a \times b.$$

$d$ is the diameter of stay over the thread, $a$ is the vertical and $b$ is the horizontal pitch.

(ii.) Longitudinal stays, 6 threads per inch, S the tensile strength in tons—

**Rule 259b.**

$$\textbf{Working pressure} = (d - 0{\cdot}340)^2 \times 9500 \times S \div 28(a \times b).$$

(iii.) Stay tubes of tested wrought iron or steel may have a working stress of 7500 lbs. at bottom of threads.

**Lloyd's old rules relating to stays.**—No steel stays are to be welded.

**Rule 260.**

Stays not exceeding $1\frac{1}{2}$ inches smallest diameter may carry, per square inch net section :—Screw stays, 8000 lbs. ; other stays, 9000 lbs.

Stays above $1\frac{1}{2}$ inches smallest diameter may carry, per square inch net section :—Screw stays, 9000 lbs. ; other stays, 10,400 lbs.

Stay tubes may carry 7500 lbs. per sq. inch of net section.

**No steel** stays may be welded. Iron stays may be welded and allowed 6000 lbs. per square inch, and unwelded above $1\frac{1}{2}$ inches diameter, 7500 lbs.

Stays other than screw stays and not exceeding $1\frac{1}{2}$ inches smallest diameter may support the same surfaces as by Board of Trade rule.

## Table CXII.—Surface of Plate supported by one Screwed Stay, 9 Threads per inch (B.M.E.D. & C. Committee Rule).

| Diameter over Threads. | Working pressures in pounds per square inch. | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 150. | 160. | 170. | 180. | 190. | 200. | 205. | 210. | 215. | 220. | 225. | 230. | 235. |
| 1¼ | 54 | 51·0 | 48·0 | 45·4 | 42·4 | 40·8 | 39·8 | 38·9 | 38·0 | 37·1 | 36·4 | 35·6 | 34·7 |
| 1⅜ | 67·7 | 63·0 | 59·7 | 56·4 | 53·4 | 50·8 | 49·6 | 48·4 | 47·3 | 46·2 | 45·2 | 44·3 | 43·5 |
| 1½ | 83·6 | 78·3 | 73·7 | 69·6 | 66·0 | 63·1 | 61·2 | 59·7 | 58·3 | 57·0 | 55·8 | 54·7 | 53·6 |
| 1⅝ | 101 | 94·7 | 89·2 | 84·2 | 79·8 | 76·3 | 73·9 | 72·3 | 70·6 | 69·0 | 67·5 | 65·8 | 65·0 |
| 1¾ | 121 | 113 | 106 | 100 | 95·4 | 91·3 | 88·4 | 86·4 | 84·4 | 82·5 | 80·7 | 78·7 | 77·6 |
| 1⅞ | 142 | 133 | 125 | 118 | 112 | 107 | 104 | 101 | 99·1 | 96·7 | 94·7 | 92·4 | 91·1 |
| 2·0 | 165 | 155 | 146 | 138 | 130 | 125 | 121 | 118 | 115 | 112 | 110 | 108 | 106 |
| 2⅛ | 190 | 178 | 167 | 158 | 150 | 143 | 139 | 136 | 133 | 129 | 126 | 124 | 122 |
| 2¼ | 215 | 202 | 190 | 180 | 171 | 163 | 158 | 154 | 151 | 148 | 145 | 141 | 138 |
| 2⅜ | 244 | 229 | 215 | 203 | 193 | 184 | 178 | 174 | 170 | 167 | 163 | 159 | 156 |

## Table CXIII.—Surface of Plate supported by one Stay of 28 tons Tensile Steel, 6 Threads per inch (B.M.E.D. & C. Committee Rule).

| Diameter over Threads. | Working pressures in pounds per square inch. | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 150. | 160. | 170. | 180. | 190. | 200. | 205. | 210. | 215. | 220. | 225. | 230. | 235. |
| 2 | 175 | 164 | 154 | 145 | 138 | 131 | 128 | 124 | 121 | 119 | 116 | 114 | 111 |
| 2⅛ | 201 | 189 | 178 | 168 | 159 | 151 | 147 | 144 | 141 | 137 | 134 | 131 | 127 |
| 2¼ | 231 | 216 | 204 | 192 | 182 | 173 | 168 | 164 | 161 | 158 | 154 | 151 | 147 |
| 2⅜ | 262 | 246 | 232 | 217 | 207 | 196 | 191 | 187 | 183 | 179 | 175 | 171 | 167 |
| 2½ | 295 | 277 | 261 | 246 | 233 | 221 | 216 | 211 | 206 | 201 | 197 | 193 | 189 |
| 2⅝ | 330 | 310 | 292 | 275 | 261 | 248 | 241 | 236 | 231 | 225 | 220 | 216 | 212 |
| 2¾ | 368 | 345 | 325 | 306 | 290 | 276 | 269 | 262 | 256 | 251 | 245 | 240 | 236 |
| 2⅞ | 406 | 382 | 360 | 339 | 321 | 305 | 297 | 290 | 284 | 278 | 271 | 266 | 261 |
| 3 | 448 | 420 | 396 | 373 | 354 | 336 | 328 | 320 | 313 | 306 | 299 | 293 | 287 |
| 3⅛ | 491 | 460 | 434 | 409 | 388 | 368 | 359 | 350 | 342 | 335 | 327 | 321 | 315 |
| 3¼ | .. | .. | 473 | 447 | 423 | 402 | 392 | 383 | 374 | 367 | 358 | 350 | 343 |
| 3⅜ | .. | .. | .. | .. | 460 | 437 | 426 | 416 | 407 | 398 | 389 | 380 | 372 |
| 3½ | .. | .. | .. | .. | .. | 474 | 462 | 451 | 441 | 431 | 421 | 412 | 403 |

For surfaces that may be supported by stays other than screw stays and exceeding $1\frac{1}{2}$ inches smallest diameter, see Table CXIII.

**The number of threads per inch** for boiler stays has been established by the Engineering Standards Committee to be 9 per inch for stays above $1\frac{1}{4}$ inches diameter, and 6 per inch for those above 2 inches, fitted with nuts on both sides of the plate.

## RULES FOR FURNACES.

**Board of Trade rules relating to furnaces.**—Cylindrical furnaces with longitudinal joints welded, or made with a butt strap double riveted, or with double butt straps single riveted,—

**Rule 261.**

$$\text{Working pressure (lbs. per sq. in.)} = \frac{99{,}000 \times (\text{thick. of plate in ins.})^2}{(\text{Length in ft.} + 1) \times \text{dia. in ins.}},$$

provided always that the pressure so found does not exceed the crushing stress per square inch of material by 4950 lbs.

**B.M.E.D. & C. Committee rules for furnaces** are as follows:—

(i.) *Plain cylindrical*, or as strengthened by the Adamson joint or other good means.

**Rule 262.**

$$\text{Working pressure} = \frac{1450(t-1)^2}{(L+24) \times D}; \text{ or} = \frac{50}{D} \times \{10(t-1) - L\}.$$

D is the external diameter and L the length between the points of support in inches; $t$ is the thickness in *thirty-seconds of an inch* and must not exceed 26. The least pressure given by the above is to be taken for use.

When the furnace is made with rings the length is to be measured between the rings.

For Fox, Morison, or Deighton furnaces, or furnaces made by J. Brown & Co., Sheffield, when they are practically true circles and not less than $\frac{5}{16}$ inch thick,—

**Rule 263. Old Board of Trade Rule.**

$$\begin{array}{l}\text{Working pressure} \\ \text{(lbs. per sq. in.)}\end{array} = \frac{14{,}000 \times \begin{array}{c}\text{thickness in inches at bottom} \\ \text{of corrugation or camber}\end{array}}{D},$$

$$\text{where } D = \begin{cases} \text{for corrugated or cambered type furnace,—outside diameter} \\ \quad \text{in inches at bottom of corrugations or cambers,} \\ \text{for ribbed and grooved furnace,—outside diameter in inches} \\ \quad \text{over plain portions.} \end{cases}$$

For bulb furnaces made by the Leeds Forge Co., the constant 14,000 may be increased to 15,000.

In all these descriptions of furnaces the plain parts at ends not to exceed 9 inches in length,—from centre of last rib to water face of back tube-plate at back, and to centre of rivets at front. Ends of plates not to be unduly thinned in flanging.

(ii.) *Corrugated, &c., Furnaces*—B.M.E.D. & C. Committee Rule.

**Rule 263a.** **Working pressure** $= C(t-1) \div D$.

$C = 480$ for the Fox, Morison, Deighton, Purves, and similar furnaces;
$C = 510$ for the Leeds Forge bulb suspension furnace;
$t$ is in *thirty-seconds of an inch*, and must not exceed 26.

**Lloyd's Old Rules for furnaces.**—The pressures that may be carried by plain cylindrical furnaces when the length of the plain part of the furnace exceeds 120 times the thickness of the plate,—

**Rule 264.**

$$\text{Working pressure (lbs. per square inch)} = \frac{1{,}075{,}200 \times T^2}{L \times D}.$$

When the length is less than 120 times,—

**Rule 265.**

$$\text{Working pressure (lbs. per square inch)} = \frac{50 \times (300T - L)}{D},$$

where $T$ = thickness of plate *in inches*;
$L$ = length of plain part in inches from rivet centre to rivet centre;
$D$ = outside diameter of furnace in inches.

**Rule 266.**

$$\text{Working pressure (lbs. per square inch)} = \frac{C \times (T-2)}{D},$$

where $T$ = thickness of plate in *sixteenths of an inch*;
$D = \begin{cases} \text{outside diameter in inches of corrugated furnaces and plain parts of Holmes' patent furnaces.} \\ \text{smallest outside diameter in inches of Brown's cambered, Improved Purves, and Leeds Forge bulb furnaces.} \end{cases}$
$C = 1259$ for steel furnaces corrugated on Fox's, Morison's, Deighton's, Beardmore's, or Leeds Forge bulb plan.
$C = 1160$ for improved Purves furnaces (ribs 9 inches apart), and Brown's cambered furnaces (ribs 8 or 9 inches apart).
$C = 912$ for spirally corrugated furnaces.
$C = 945$ for Holmes' patent furnaces, when corrugations are not more than 16 inches pitch, nor less than 2 inches high.

## RULES RELATING TO TESTING BOILERS GENERALLY.

**Hydraulic tests.**—New boilers should be tested by hydraulic pressure before being lagged or placed in the vessel. The test pressure to remain on for at least ten consecutive minutes in all cases.

**Board of Trade, Lloyd's, and other Registries.**—Twice the working pressure for boilers whose working pressure does not exceed 100 lbs. Over 100 lbs. W. P. the test is 1·5 × W.P. + 50 lbs.

**British Admiralty.**—W.P. + 90 lbs.

**German Authorities.**—W.P. + 75 lbs.

**French Authorities.**—W.P. + 85 lbs.

Boilers which have been in use are not to exceed 1·5 × W.P.

### British Mercantile Hydraulic Tests of New Marine Boilers.

30. In all new boilers working at pressures up to 100 lbs. per square inch the hydraulic test must be twice the working pressure. For boilers working at pressures greater than 100 lbs. per square inch the hydraulic test pressure must be 1½ times the working pressure plus 50 lbs. per square inch.

### Test of Boilers which have been in Service.

31. In boilers which have been in service the hydraulic test when applied should not exceed 1½ times the working pressure.

## EVAPORATORS, &c. (Board of Trade Rules).

164. Evaporators, generators, feed make-ups, &c., where the evaporation of water under pressure is an essential feature, should be regarded as steam boilers, whether the evaporation is effected by heat from coal gas, from steam, or from any other source, and they should be examined by the Surveyor on each occasion the vessel is surveyed for passenger certificate in the same manner as other boilers on board the vessel; and the particulars regarding them, their safety-valves, &c., should be recorded on the declaration in the same manner as is done in the case of other auxiliary boilers.

The strength, quality of material, and method of construction of such apparatus, should, as a rule, be in accordance with the regulations for steam boilers. For moderate pressures, however, evaporators of the type usually fitted may be made of cast material, but in no case should the pressure exceed 15 lbs. per square inch when the main body of the apparatus is a single casting. Subject to this limitation and to the sanction of the Board for the use of such material in each particular case, evaporators made of cast-iron, or of gun-metal, having a tensile strength of not less than 10 tons per square inch, may be allowed a working pressure not exceeding that found by the following formulæ, provided the thickness is not less than $\frac{5}{8}$ inch in the case of cast-iron, and $\frac{3}{8}$ inch in the case of gun-metal, and the castings are in every way sound and to the Surveyor's satisfaction :—

### Cylindrical Shells.

$$\frac{C(T-\frac{1}{8})}{D} = \text{working pressure.}$$

### Circular Flat Surfaces.

$$\frac{C_1 \times T^2}{D^2} = \text{working pressure.}$$

### Square Flat Surfaces.

$$\frac{C_2 \times T^2}{S^2} = \text{working pressure.}$$

Where $T$ = thickness, in inches ;
$D$ = diameter, in inches ;
$S$ = side, in inches ;
$C$ for cast-iron = 4,000 ;
$C_1$ ,, ,, ,, = 24,000 ;
$C_2$ ,, ,, ,, = 16,000 ;
$C$ ,, gun-metal = 6,000 ;
$C_1$ ,, ,, ,, = 30,000 ; and
$C_2$ ,, ,, ,, = 20,000.

If cast-steel is used, the minimum thickness should not be less than $\frac{1}{2}$ inch, and the constants $C$, $C_1$, and $C_2$ may be 10,400, 52,000, and 34,700 respectively.

The formulæ may also be used for determining the working pressure permissible for feed heaters, feed filters, &c., and for such vessels the constants for cast-iron and cast-steel may be increased by 25 per cent.

Where there are large branches, doors, or other large openings in the castings, the scantlings will require to be materially increased, and such cases should be submitted for consideration before being passed.

When the ends are cast solid with the shell, there should be a substantial fillet all round inside, and when the ends are bolted, D, in the flat surface formula, should be the diameter of the bolt circle. The flange should be of sufficient thickness and have a substantial fillet at the root all round.

165. In calculating the strength of studs or bolts securing the covers of evaporators, feed heaters, feed filters, and other similar vessels, the

calculated load on the cover should be found by multiplying the area of the pitch circle by the working pressure, and the nominal stress allowed on the net section of the material of the studs, &c., should not exceed 7000 lbs. per square inch in the case of iron, and 9000 lbs. per square inch in the case of steel, the maximum allowance being permissible only when the diameter is $\frac{7}{8}$ inch or more. When the studs or bolts are of a diameter less than $\frac{7}{8}$ inch, the nominal stress allowed per square inch of net section should not exceed 6000 lbs. in the case of iron, and 7200 lbs. in the case of steel, owing to the relatively greater stress to which they are subject when the nuts are tightened.

Studs or bolts securing covers which are required to be frequently removed should not be less than $\frac{7}{8}$ inch in diameter.

**166.** The mountings, &c., for evaporators should, as a general rule, be similar to those required in the case of boilers on board passenger vessels, but a single safety-valve may be allowed, provided it is of sufficient size.

**167.** When a reducing nozzle is fitted in the steam supply pipe, the contracted orifice should not, in ordinary cases, exceed that found by the following formula:—

$$\frac{A \times p}{6 \times P} = \text{area of orifice,}$$

where A = combined area of safety-valves fitted to the evaporator;
$p$ = absolute pressure at which the evaporator is worked; and
P = absolute pressure of entering steam.

The reduced orifice for an evaporator having only one safety-valve should not, in any case, be greater than would be allowed, for the same pressures, with a single valve $2\frac{13}{16}$ inches in diameter (*i.e.* equivalent in area to two 2-inch valves).

Reducing orifices should be bored through brass or similar material, and should be parallel for a length of at least $\frac{1}{4}$ inch; and each nozzle should be formed with a facing at the side, on which particulars regarding the safety-valves, their load, the maximum pressure of the entering steam, and the diameter of the orifice should be stamped, as shown by the following example:—

2 S.V. Dia. 3″. Load 10 lbs. sq. in.
Boiler Press. 160 lbs.
Reduced Orifice 21/32″ Dia.

**168.** On the completion of the hydraulic test of an evaporator, a feed heater, feed filter, or other similar vessel which complies with the Board's regulations and has been surveyed during construction, the Surveyor, in order to afford means of identification, should stamp the apparatus in a conspicuous place, with the pressure applied to the shell, the pressure applied to the coils (if any), the date, and his initials.

169. Feed pipes, feed heaters, filters or other vessels through which the feed water passes on its way from the pumps to the boilers, should be made sufficient for a pressure 20 per cent. in excess of the boiler pressure; and an efficient relief-valve (or valves), suitably adjusted, and of a type which does not present a ready means of overloading, should be fitted where required to prevent this pressure being exceeded in any part of the feed system, under any conditions likely to occur. In cases where the main feed pumps are independent of the main engines, the area of the relief-valve (or valves) should not, as a rule, be less than half that of the feed discharge pipe, provided the latter is sufficient to ensure a reasonable velocity of flow in the pipe; but, when the main pumps are worked by the main engines, the relief-valve (or valves) should be of at least the same area as the pipe.

**Local heating** of plates should be avoided.

**Minimum thickness** of plates to be used, $\frac{5}{16}$ inch.

**Annealing.**—All plates which have been punched, flanged, or locally heated, and all stays and stay tubes which have been locally heated, must be carefully annealed afterwards.

**Welding.**—Steel plates that have been welded should not be passed to carry a tensile stress; when subject to a compressive stress they should be efficiently annealed after welding.

**Boiler tracings, &c.**—Tracings of boilers may be received for examination upon payment of the usual fee of £2, and the Surveyors may proceed as far as witnessing the hydraulic test before any further instalment of the survey fee is paid. Engineers and boilermakers should be advised of this arrangement.

**Donkey boilers** that are in any way attached to, or connected with, the main boilers, or with the machinery used for propelling the vessel, should be surveyed and have their working pressure fixed in the same way as the main boilers, and have a water and steam gauge, and all other fittings complete, and, as regards safety valves, should comply with the same regulations as the main boilers.

**Launch boilers.**—The boilers of steam launches forming part of the statutory boat capacity of passenger steamers should as regards construction, strength, material, safety valves, and other fittings comply with the same regulations as the main boilers.

## LLOYD'S RULES RELATING TO STEEL BOILERS GENERALLY.

### Boilers.

**Section 3.** 1. The Surveyors will be guided in fixing the working pressure by the tables and formulæ annexed. (*See* section 11.)

2. Any novelty in the construction of the machinery or boilers to be reported to the Committee.

3. The boilers, together with the machinery, to be inspected at different stages of construction.

All the holes in steel boilers should be drilled, but if they be punched the plates are to be afterwards annealed.

All plates that are dished or flanged, or in any way heated in the fire for working, except those that are subjected to a compressive stress only, are to be annealed after the operations are completed.

No steel stays are to be welded.

Unless otherwise specified, the Rules for the construction of iron boilers will apply equally to boilers made of steel.

4. The boilers to be tested by hydraulic pressure, in the presence of the Engineer-Surveyor, to twice the working pressure, and carefully gauged while under test.

5. Two safety valves to be fitted to each boiler, and loaded to the working pressure in the presence of the Surveyor. In the case of boilers of greater working pressure than 60 lbs. per square inch, the safety valves may be loaded to 5 lbs. above the working pressure. If common valves are used, their combined areas to be at least half a square inch to each square foot of grate surface. If improved valves are used they are to be tested under steam in the presence of the Surveyor; the accumulation in no case to exceed 10 per cent. of the working pressure.

6. An approved safety valve also to be fitted to the superheater.

7. In winch boilers one safety valve will be allowed, provided its area be not less than half a square inch per square foot of grate surface.

8. Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing gear which must lift the valve itself. All safety-valve spindles to extend through the covers and be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

9. Stop-valves to be fitted so that each boiler can be worked separately.

10. Each boiler to be fitted with a separate steam gauge, to accurately indicate the pressure.

11. Each boiler to be fitted with a blow-off cock independent of that on the vessel's outside plating.

12. The machinery and boilers are to be securely fixed to the vessel to the satisfaction of the Surveyor.

## RIVETED JOINTS.

**Board of Trade.**—The following sketches of riveted joints, and formulæ for determining their various proportions, are given in an appendix to the Board of Trade Rules; the formulæ are here given in a form differing slightly from that adopted in the Rules:—

F in the following formulæ stands for the factor of safety (for which see Table CIV.), $r$ for percentage of plate left between rivet holes, and $S_1$ for tensile strength of plates in tons.

### ORDINARY CHAIN AND ZIGZAG RIVETED JOINTS.

$$\left.\begin{array}{r}\text{Percentage of plate}\\ \text{left between holes}\end{array}\right\} = \frac{100\,(\text{pitch} - \text{diameter of rivet})}{\text{pitch}} = r.$$

$$\left.\begin{array}{l}\text{Nominal per-}\\ \text{centage of}\\ \text{rivet section}\end{array}\right\} = \frac{\left.\begin{array}{l}958{\cdot}33\ \text{Butt}\\ 511{\cdot}11\ \text{Lap}\end{array}\right\} \times \text{area of riv.} \times \text{No. rivets in pitch} \times \text{F}}{\text{S}_1 \times \text{pitch} \times \text{thickness of plate}}.$$

**Rule 267.** To find pitch so that nominal rivet section and net plate section may be of equal strength:—

$$\text{Pitch} = \frac{\left.\begin{array}{l}958{\cdot}33\ \ \text{Butt}\\ 5{\cdot}111\ \text{Lap}\end{array}\right\} \times \text{area rivet} \times \text{No. rivs. in pitch} \times \text{F}}{\text{S}_1 \times \text{thickness of plate}} + \text{dia. rivet}.$$

**Rule 267a.** To find pitch and diameter of rivet:—

$$\left.\begin{array}{r}\text{Diameter}\\ \text{of rivet}\end{array}\right\} = \frac{\text{S}_1 \times r \times \text{thickness of plate}}{\left.\begin{array}{l}7{\cdot}526\ \text{Butt}\\ 4{\cdot}014\ \text{Lap}\end{array}\right\} \times (100 - r) \times \text{No. rivets in pitch} \times \text{F}}.$$

$$\text{Pitch} = \frac{\left.\begin{array}{l}13{\cdot}29\ \ \text{Butt}\\ 24{\cdot}911\ \text{Lap}\end{array}\right\} \times \text{S}_1 \times r \times \text{thickness of plate}}{(100 - r)^2 \times \text{No. rivets in pitch} \times \text{F}}.$$

Also, when diameter of rivet is found first:—

**Rule 268.** $\text{Pitch} = \dfrac{100 \times \text{diameter of rivet}}{100 - r}.$

When double butt straps are used each strap must have a thickness of $\frac{5}{8}$ths of the plate it covers.

A single butt strap must have a thickness equal to $1\frac{1}{8} \times$ thickness of plate it covers.

**Distance from centre of rivet to edge of plate (joint E) = $1\frac{1}{2} \times$** diameter of rivet.

**Distance between rows of rivets:—**

(*a*) **Chain riveted joints** (figs. B, C, $\text{E}_2$, G),—

**Rule 269.**

$\text{V} =$ not less than $2 \times$ dia. of rivet, preferably $\dfrac{(4\ \text{dia. rivet}) + 1}{2}$.

(*b*) **Zigzag riveted joints** (figs. $\text{B}_1$, $\text{C}_1$, $\text{E}_1$, $\text{G}_1$),—

$$\text{V} = \frac{\sqrt{(11\ \text{pitch} + 4\ \text{dia. rivet})\,(\text{pitch} + 4\ \text{dia. rivet})}}{10}.$$

Diagonal pitch (figs. $B_1$, $C_1$, $E_1$, $G_1$),—

$$p_D = \frac{6 \text{ pitch} + 4 \text{ dia. rivet}}{10}.$$

## RIVETED JOINTS WITH ALTERNATE RIVETS IN OUTER, OR OUTER AND INNER, ROWS LEFT OUT.

$$\text{Percentage of plate left between holes} = \frac{100\,(\text{pitch} - \text{dia. of rivet})}{\text{pitch}}$$

$$\left.\begin{array}{l}\text{Nominal per-}\\ \text{centage of}\\ \text{rivet section}\end{array}\right\} = \frac{\left.\begin{array}{l}958{\cdot}33 \text{ butt}\\ 511{\cdot}11 \text{ lap}\end{array}\right\} \times \text{area of riv.} \times \text{No. rivets in pitch} \times F}{S_1 \times \text{pitch} \times \text{thickness of plate}}.$$

Double butt straps for this type of joint must each be of thickness given by,—

**Rule 270.**

$$\text{Thickness of butt strap} = \frac{5 \times \text{thick. of plate} \times (\text{pitch} - \text{dia. of rivet})}{8 \times (\text{pitch} - 2 \text{ dia. of rivet})}.$$

Distance from centre of rivet to plate edge (joint E, p. 385) = $1\frac{1}{2}$ × diameter of rivet.

**Rule 271.** Distance between rows of rivets :—

(*a*) Chain riveted joints (H, p. 385),—

$$\left.\begin{array}{l} V = \dfrac{\sqrt{(11 \text{ pitch} + 4 \text{ dia. rivet})\,(\text{pitch} + 4 \text{ dia. rivet})}}{10} \\[2ex] \text{or } V = \text{not less than } 2 \times \text{dia. of rivet, and preferably } \dfrac{(4 \times \text{dia. rivet}) + 1}{2} \end{array}\right\} \begin{array}{l}\text{The greater}\\ \text{of these}\\ \text{two values}\\ \text{to be used.}\end{array}$$

(*b*) For joint H (p. 385),—

$V_1 = 2$ × dia. of rivet as a minimum, but the value $\dfrac{(4 \times \text{dia. of rivet}) + 1}{2}$ is preferable.

(*c*) Zigzag riveted joints ($H_1$, p. 385),—

$$V = \sqrt{(\tfrac{11}{20} \text{ pitch} + \text{dia. rivet})\,(\tfrac{1}{20} \text{ pitch} + \text{dia. rivet})}.$$

Diagonal pitches :—

(*a*) Diagonal pitch ($H_1$, p. 385),

$$p_D = \tfrac{3}{10} \text{ pitch} + \text{dia. of rivet}.$$

(*b*) For joint $H_1$ (p. 385),—

$$P_D = \frac{3 \text{ pitch} + 4 \text{ dia. rivet}}{10}.$$

Distance between inner rows of rivets (joint $H_1$, p. 385),—

$$V_1 = \frac{\sqrt{(11 \text{ pitch} + 8 \text{ dia. rivet})(\text{pitch} + 8 \text{ dia. rivet})}}{20}.$$

## Maximum Pitches for Riveted Joints.

T = thickness of plate in inches ;

$p$ = maximum pitch of rivets in inches, provided it does not exceed $12\frac{1}{4}$ inches ; and

C = constant applicable from the following table :—

| Number of Rivets in one Pitch. | Constants for Lap Joints. | Constants for Double Butt-Strap Joints. |
|---|---|---|
| 1 | 1·31 | 1·75 |
| 2 | 2·62 | 3·50 |
| 3 | 3·47 | 4·63 |
| 4 | 4·14 | 5·52 |
| 5 | ... | 6·00 |

$$(C \times T) + 1 = p.$$

## Diagrams of Board of Trade Rules.

Fig. (A)

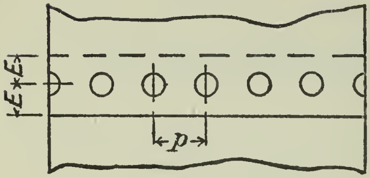


Fig. (B)

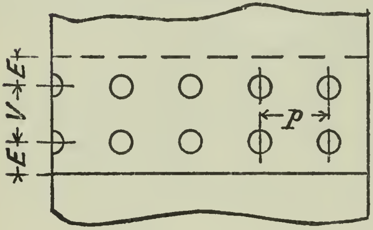


Fig. (B₁)

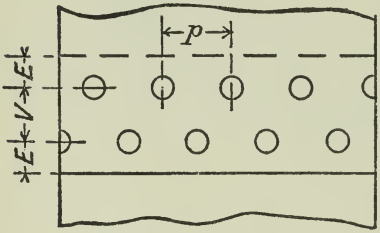


Fig. (C)

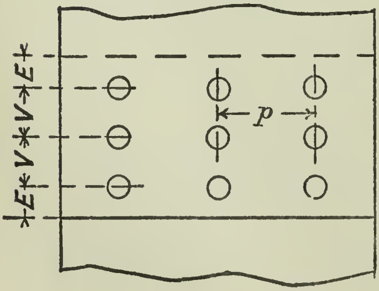


Fig. ($C_1$)

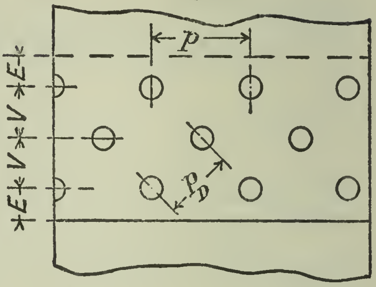


Fig. (D)

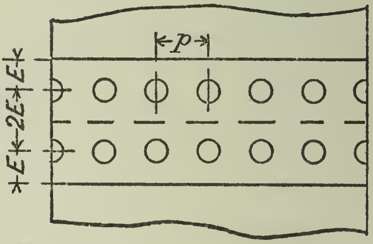
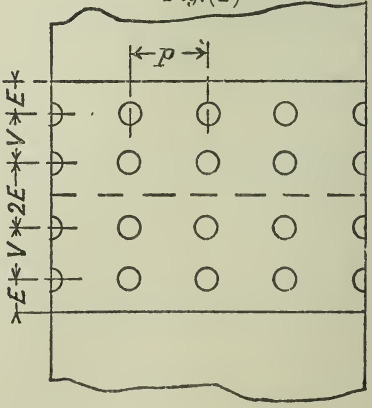
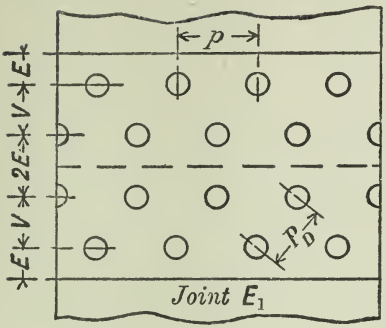
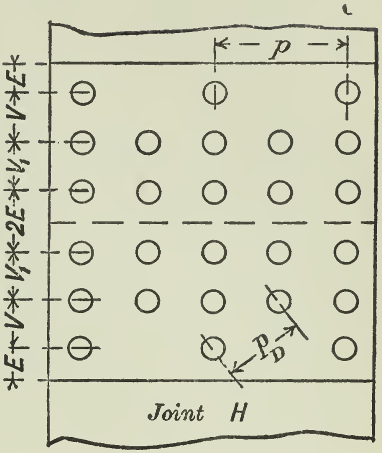


Fig. (E)

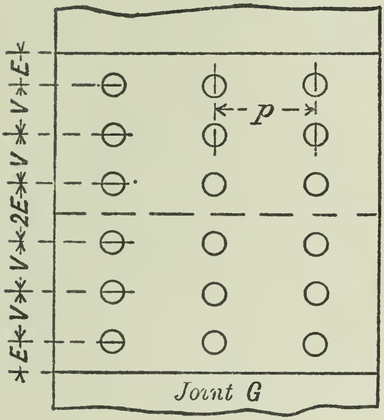




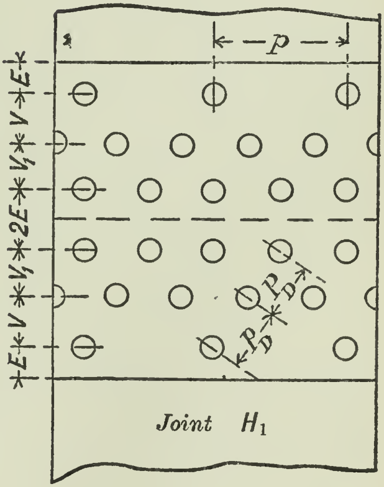
Joint $E_1$



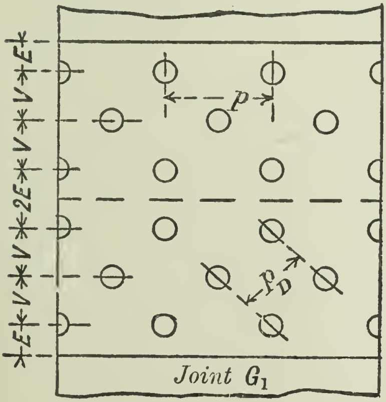
Joint H



Joint G

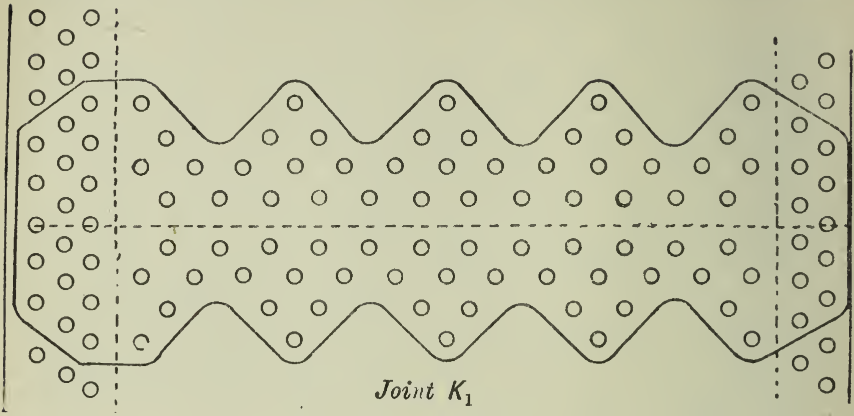


Joint $H_1$

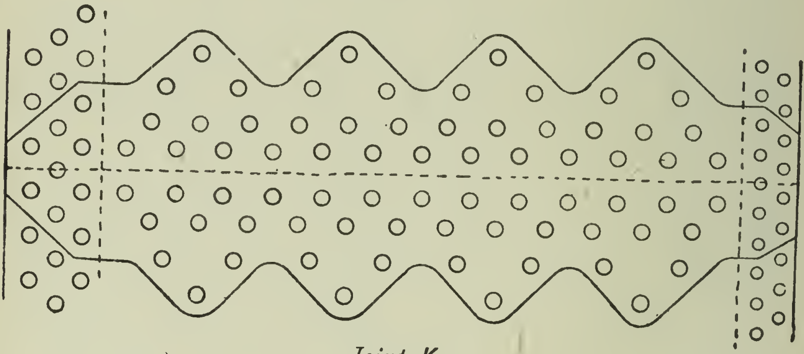


Joint $G_1$

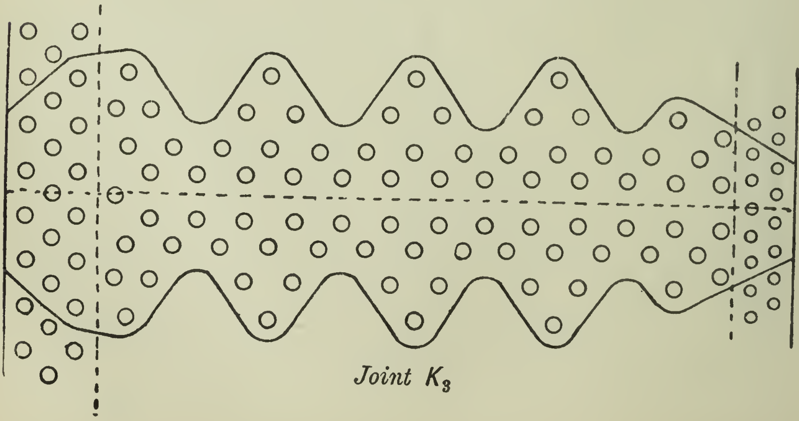
The following figs. are examples of quadruple riveting of the special kinds adopted largely on the Continent for boilers having thick shells.



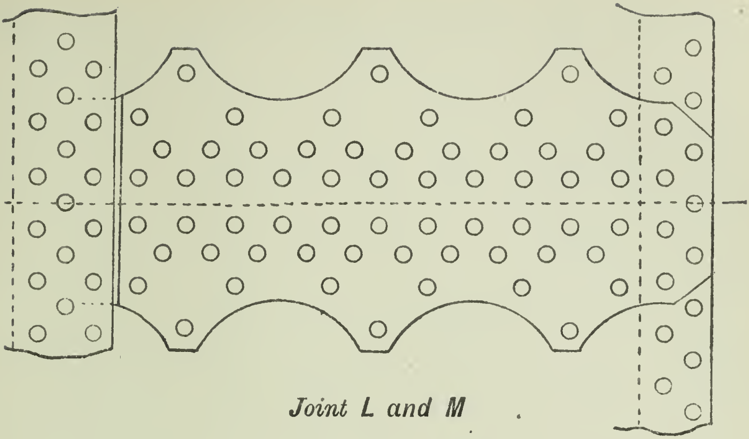
*Joint* $K_1$



*Joint* $K_2$

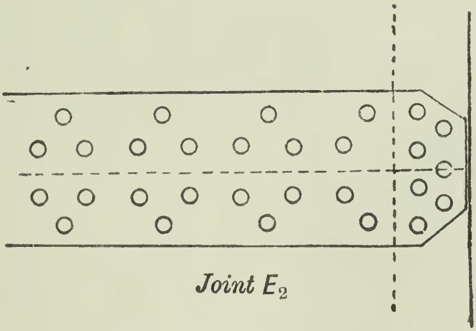


*Joint* $K_3$



*Joint L and M*

Joint $E_2$ is an example of a double riveted joint with alternate rivets outer rows omitted :—



*Joint $E_2$*

## Table CXIV.—Tests of Boiler Material (Steel).

| Part of Boiler. | Admiralty. | | Board of Trade. | | Lloyd's Register. | | British Corporation. | | Bureau Veritas. | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Ultimate Strength. | Extension. | Ultimate Strength. | Extension. | Ultimate Strength. | Extension. | Ultimate Strength. | Extension. | Ultimate Strength. | Extension. |
| | | Per cent. | | Per cent. | | Per cent. | | Per cent. | | Per cent. |
| Shell plates, Tons per sq. in. | 27 to 30 | 20 | 27 to 32 | 18 | 27 to 32 | 20 | 27 to 32 | 20 | to 30 | 20 |
| Plates exposed to flame, „ | 24 to 27 | 25 | 26 to 30 | 20 | 26 to 30 | 20 | 25 to 30 | 20 | to 28 | 22 |
| Rivet bars, „ | 24 to 27 | 25 | 26 to 30 | 25 | 26 to 30 | 20 | 25 to 30 | 20 | 24 to 27 | 25 |
| Stay bars, „ | 27 to 30 | 20 | 27 to 32 | 20 | 26 to 30 | 20 | 27 to 32 | 20 | to 30 | 20 |
| Steam-pipe plates, „ | 24 to 27 | 33 | .. | .. | 26 to 30 | 20 | .. | .. | .. | .. |
| Corrugated furnaces, „ | 23 to 25 | 27 | .. | .. | 26 to 30 | 20 | 25 to 29 | 20 | to 25 | 25 |
| Tubes, . . . . | 26 | 27 | 26 to 30 | 20 | .. | .. | .. | .. | .. | .. |

**Table CXV.—Various Kinds of Jointings for Plates and their Relative Values when designed in accordance with Board of Trade Rules.**

| Mark. | Description of Joint. | Equal to per cent. of Solid Plate. | Relative Working Pressure. |
|---|---|---|---|
| A | Lap joint, single riveted . . . | 56 | 105 |
| B | ,, double ,, . . . | 66 | 126 |
| C | ,, treble ,, . . . | 72 | 140 |
| D | Butt joint, with double butt straps, single riveted . . . . . | 66 | 126 |
| E | Butt joint, with double butt straps, double riveted . . . . . | 75 | 150 |
| $E_2$ | Butt joint, double straps, double riveted, half number in outer rows . | 81/84 | 162/168 |
| G | Butt joint, double straps, treble riveted . . . . . . . | 80/82 | 160/164 |
| H | Butt joint, double straps, treble riveted, half number in outer rows . | 84/88 | 168/176 |
| J | Butt joint, double straps, quadruple riveted . . . . . . | 84 | 168 |
| K | Butt joint, double straps, quadruple riveted, special, 9 to the pitch . . | 91·8 | 183·6 |
| L | Butt joint, double straps, quadruple riveted, special, 11 to the pitch . | 93·2 | 186·4 |
| M | Butt joint, double straps, quadruple riveted, special, 11 to the pitch . | 94·2 | 188·4 |

*N.B.*—In cases $E_2$, G, and H the higher percentage is obtained by using rivets large in diameter compared with the thickness of plates. Case $E_2$ is only suitable for plates comparatively thin, say up to $\frac{7}{8}$ inch, as the rivets have to be $1\frac{1}{2}$ times the thickness for steel of 27 tons tensile, and consequently, if 36-ton steel is used, the diameter will be excessive and the pitch too great for tight work.

In case M the very high percentage is obtained by using rivets of diameter 1·125 times the thickness of the 27-ton plate, as against the simple $d=t$ for 93·2 per cent.

## SUPERVISION OF BOILER WORK.

**Admiralty.**—The following instructions to boilermaker overseers are those that were given in Admiralty specifications for cylindrical boilers:—

The boilers will be subject to the supervision of an overseer, who will be directed to attend on the premises of the contractors during the progress of the work on the boilers, to examine the material and workmanship used in their construction, to witness the prescribed tests, and to see that this specification, as regards the boilers and work in connection, is conformed to in all respects by the contractors. The extent of supervision is described in the following paragraphs extracted from Admiralty instructions to overseers, and the contractors are to afford him every facility for their proper execution.

The plates and other material used in the construction of the boilers to be subjected to such tests as may be directed in the specification. Every plate used is to be carefully examined by the overseer for laminations, blisters, veins, and other defects, and to ensure that it is of the proper thickness and brand. No plate, angle, &c., which from any cause is considered by the overseer to be unfit for the intended use is to be fitted.

During the construction of the various parts of the boilers, the overseer is to satisfy himself that the dimensions as shown on the approved drawings are being adhered to by the contractors.

Whenever plates are flanged or welded, or in any case where iron or steel is worked in such manner that it is particularly liable to suffer in strength unless carefully handled, the overseer is to be present if possible on all occasions during the time the work on each article is in progress, and he is to fully satisfy himself that it is sound before he allows any part to be put in the boilers.

Samples of the rivets being used for the boilers are to be taken by the overseer during the progress of the work and tested as specified hereafter, and any batches of rivets found defective are to be rejected. Before rivets are put in, the overseer is to see that the plates are brought properly together, and that the holes are fair with one another. He is not to allow drifting on any account, but he is to see that they are carefully rimed fair where necessary. He is also to make sure during the progress of the work that the rivets fill the holes completely, and that the heads are properly set up, well formed, and finished.

The overseer is to see that all internal parts of the boiler are riveted with rivets having heads and points of approved shape, and that any seams he considers necessary are riveted on the fire side. No snap heads are to be allowed in the internal parts. Any proposal for hydraulic riveting the internal parts is to be submitted to the Admiralty, with sketch of the proposed heads and points. In all parts where the rivets are not closed by hydraulic riveting machinery, he is to see that the rivet holes are countersunk and that coned rivets are used. All holes in the plates, angles, &c., are to be drilled, and not punched, and are to be drilled in place after bending. The clearance between rivet hole and rivet before closing is not to be greater than approved by the overseer.

The overseer will see that the particulars of the form, dimensions, and pitch of the various stays shown on the drawings are adhered to, and samples of them are to be tested as directed in this specification; and he will be guided by his experience as a workman in testing and judging of the soundness of the forging and construction of the various stays.

He is to see that palm stays if fitted are forged from the solid and not welded, that all short stays are nutted on all flat surfaces except where otherwise approved and screwed to a pitch of eight threads per inch for stays of 1 inch diameter and above, that the holes for the screwed stays in the water spaces are drilled and tapped together after the furnaces and combustion chambers have been riveted in place in the boiler, that the combustion-chamber stays are drilled square to the bevel of the combustion-chamber plates, and that no bevel washers are inside the chamber. Any girder stays used for combustion chambers are to be well bedded on to the tube plates to the satisfaction of the overseer.

The overseer is to see that the arrangement of the zinc plates shown on the approved drawings is adhered to, that the metallic surfaces in contact are filed bright, and that means are adopted to secure a firm grip of the clips by which the plates are attached.

The overseer is to witness the testing, in all cases, of the boiler tubes, in accordance with this specification, before they are put in the boiler.

When the boilers are reported to the overseer by the contractors as being completed, ready for testing by water pressure, the overseer is to witness a preliminary test of them in accordance with the specification, carefully observing with the assistance of gauges and straight edges whether any bulging or deflection of the plates has taken place.

The official test will be conducted on all occasions in the presence of an inspecting officer. A test pressure gauge is supplied to the overseer from the Admiralty, and the official test is to be made with this gauge.

After the boilers have been tested by water pressure the overseer is to see that they are properly cleaned inside and outside, and then well painted outside with red lead. It is important that the whole surface of the boilers should be thoroughly cleansed of scale formed in manufacture before any paint is put on them. The boilers are not to be exposed to the weather till they are so painted, and properly cleaned and closed up to his satisfaction.

The overseer is to make himself fully acquainted with the progress of the whole of the work in its various stages, to satisfy himself that every part is sound before it is allowed to be put in the boilers, and to see that the following instructions for the treatment of mild steel are strictly complied with.

**Treatment of mild steel.**—All plates or bars which can be bent *cold* are to be so treated; and if the whole length cannot be bent cold, heating is to be had recourse to over as little length as possible.

All plates of the boilers are to be flanged by hydraulic pressure, and in as few heats as possible.

In cases where plates or bars have to be heated, the greatest care should be taken to prevent any work being done upon the material after it has fallen to the dangerous limit of temperature known as a "blue heat," say from 600° to 400° F. Should this limit be reached during working, the plates or bars should be reheated.

Plates or bars which have been worked locally while hot are to be subsequently annealed over the whole of each plate or bar.

All plates for boilers and steam pipes and all tubes are to be treated as follows for removal of scale:—Previous to work being commenced on them, they are to stand for eight hours in a mixture of 19 of water to 1 of hydrochloric acid. They should be placed on edge and not laid flat. On removal from acid bath, they should be thoroughly brushed and washed in fresh water, and then placed on edge to dry.

*N.B.*—A shorter time in a somewhat stronger solution is safer, as it precludes the occluding of hydrogen and thereby making the metal brittle.

## BOILER MOUNTINGS, &c.

**Size of main stop-valves** should be such that the whole of the steam is just capable of passing through without appreciable loss of pressure and no more. The lift of valve should be no more than will permit of this, for then, if with careless stoking or other causes the steam drop, it tends to cause priming. It is surprising how small a hole in a diaphragm, interposed between the steam-pipe flanges, will pass all the steam a boiler of large size can make. Such a diaphragm may be used often with advantage on boilers which are in the hands of careless or unskilled engineers, as it cannot be tampered with as can be the stop-valve.

The flow of steam through the lengths of pipe on ship-board may be taken as 6000 feet per minute in pipes under 2·5 inches diameter, 7000 up to 4 inches, 8000 up to 7 inches, 9000 to 12 inches and over 12 inches, 9500 to 15 inches per pressure about 700 lbs. per square inch. That is, velocity of boiler steam $= 5000 \times \sqrt[4]{\text{diameter}}$.

**Rule 272.** **Area steam-pipe section** $= \dfrac{\text{T.H. surf.}}{p} \times f + 3$ sq. in.,

for turbines, $f = 1·25$; for quick-revolution reciprocators, 1·55; for ordinary mercantile marine reciprocators, 1·75.

**Rule 273.** **Area of steam-pipe section** $= \dfrac{\text{I.H.P.}}{p} \times \dfrac{c}{x} + 3$ sq. ins.,

$c$ is the consumption of steam per I.H.P. hour, as given in Table XCIII.; $x$ is $= 6·2$ for turbines; $x = 5·1$ for quick-revolution reciprocators, and $x = 4·3$ for slow-running mercantile engines.

The area of stop-valve should be not less than 10 per cent. greater than the pipe to provide equal area clear of obstructions, that is,

**Rule 274.** **Diameter of stop-valve** = 1·06 × diameter of steam-pipe. If in certain cases the boilers are going to be forced so that the production is more than 8 lbs. per square foot of total heating surface, the factor $f$ must be increased proportionately.

The following Tables will be found useful as a guide in determining the size of steam-piping from the demand requirements.

## Table CXVI.—Quantity of Steam delivered per minute through Pipes whose length does not exceed $240 \times \sqrt{\text{diameter}}$ under Service Conditions.

| Diameter of Bore. | Live Steam, 200 lbs. Absolute. | | Exhaust Steam, 3 lbs. Absolute. | | Diameter of Bore. | Live Steam, 200 lbs. Absolute. | | Exhaust Steam, 3 lbs. Absolute. | |
|---|---|---|---|---|---|---|---|---|---|
| | Cubic Feet. | Lbs. | Cubic Feet. | Lbs. | | Cubic Feet. | Lbs. | Cubic Feet. | Lbs. |
| ins. | | | | | ins. | | | | |
| 1·0 | 27·19 | 11·71 | 17·95 | 0·152 | 9·0 | 3826 | 1649 | 2525 | 21·40 |
| 1·5 | 69·51 | 30·00 | 45·94 | 0·389 | 10·0 | 4852 | 2100 | 3202 | 27·14 |
| 2·0 | 129·7 | 54·12 | 75·60 | 0·646 | 12·0 | 7300 | 3144 | 4818 | 40·83 |
| 2·5 | 213·9 | 92·12 | 141·2 | 1·196 | 14·0 | 10,357 | 4012 | 6836 | 57·91 |
| 3·0 | 322·6 | 138·9 | 212·9 | 1·804 | 16·0 | 13,950 | 6000 | 9207 | 78·00 |
| 3·5 | 456·3 | 196·6 | 301·2 | 2·552 | 18·0 | 18,018 | 7760 | 12,000 | 101·7 |
| 4·0 | 616·8 | 265·6 | 407·7 | 3·456 | 20·0 | 23,050 | 9930 | 15,220 | 129·0 |
| 5·0 | 1019 | 438·9 | 672·5 | 5·700 | 25·0 | ... | ... | 25,210 | 213·7 |
| 6·0 | 1536 | 661·5 | 1014 | 8·590 | 30·0 | ... | ... | 38,000 | 321·8 |
| 7·0 | 2180 | 939·0 | 1419 | 12·02 | 35·0 | ... | ... | 53,600 | 453·6 |
| 8·0 | 2938 | 1263 | 1939 | 16·43 | 40·0 | ... | ... | 71,800 | 608·5 |

*N.B.*—For other pressures—*e.g.* 150 lbs. absolute, multiply by 0·75 ; 180 lbs., by 0·9 ; 215 lbs., by 1·075 ; 225 lbs., by 1·125 ; 240 lbs., by 1·2.

For the flow of superheated steam, *vide* Table LXXVa., p. 261.

### Table CXVII.—Pounds of Steam delivered per hour through Pipes with loss of 1 lb. per 100 feet.

| Bore of Pipe. | Pressure of steam absolute in lbs. per square inch. | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | 5 | 10 | 20 | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 225 | 250 |
| ins. | | | | | | | | | | | | | |
| 1 | ... | ... | ... | ... | 104 | 123 | 144 | 160 | 174 | 187 | 200 | 212 | 222 |
| 2 | ... | ... | ... | 382 | 590 | 718 | 815 | 903 | 984 | 1,060 | 1,130 | 1,198 | 1,265 |
| 3 | 356 | 549 | 760 | 1,054 | 1,620 | 1,975 | 2,245 | 2,490 | 2,715 | 2,925 | 3,120 | 3,310 | 3,450 |
| 4 | 750 | 1,140 | 1,550 | 2,085 | 3,120 | 3,850 | 4,500 | 5,060 | 5,600 | 6,100 | 6,600 | 7,050 | 7,500 |
| 5 | 1,300 | 1,970 | 2,700 | 3,610 | 5,115 | 6,650 | 7,900 | 8,960 | 9,910 | 10,720 | 11,450 | 12,100 | 12,700 |
| 6 | 2,015 | 3,100 | 4,290 | 5,960 | 9,170 | 11,000 | 12,700 | 14,000 | 15,350 | 16,500 | 17,640 | 18,600 | 19,500 |
| 7 | 3,010 | 4,580 | 6,320 | 8,800 | 13,600 | 16,500 | 19,000 | 21,050 | 22,900 | 24,500 | 26,000 | 27,400 | 28,700 |
| 8 | 4,200 | 6,730 | 8,830 | 12,300 | 19,200 | 23,050 | 26,300 | 29,100 | 31,800 | 34,050 | 36,150 | 38,200 | 40,100 |
| 9 | 5,550 | 8,550 | 11,820 | 16,400 | 25,300 | 30,500 | 35,000 | 38,900 | 42,300 | 45,500 | 48,600 | 51,300 | 54,000 |
| 12 | 11,380 | 17,500 | 24,300 | 33,700 | 51,800 | 62,800 | 71,700 | 79,500 | 86,600 | 93,300 | 99,600 | 106,000 | 112,000 |
| 15 | 19,900 | 30,610 | 42,400 | 58,900 | 90,500 | ... | ... | ... | ... | ... | ... | ... | ... |
| 18 | 31,400 | 48,400 | 67,000 | 93,000 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 21 | 46,100 | 68,300 | 99,500 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 24 | 64,400 | 90,200 | 137,000 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 27 | 85,000 | 114,000 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 30 | 105,000 | 130,000 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |

*N.B.*—This table is based on the assumption that the only resistance to flow arises from the friction of the projecting bends; valves, branches, and other such things found in practice add to the resistance and therefore cause a diminution of flow, or a greater reduction in pressure for the same flow.

**Stop-valves.**—The diameter of the boiler stop-valves is often fixed from the previously determined size of main steam pipe at engines. When the branch pipes from two or more separate but similar boilers join together into one main steam pipe, of diameter D, the size of each branch pipe (their number being $n$) may be that given by,—

**Rule 275.—Diameter of Branch pipe to each boiler** $= \mathrm{D}\sqrt{\frac{4}{3n}}$.

The more important points connected with the design and construction of stop-valves have already been dealt with (pages 286 and 392), and it is only necessary to add here that boiler stop-valves should have turned spigots fitting accurately into the holes in the boiler plates,—which should be carefully cut out by means of a bar and cutter, and *not* by hammer and chisel,—and that the flanges and necks should be extra strong and well ribbed, and the bolts or studs (bolts with heads inside the boiler are best) by which they are attached to the boiler, larger and more numerous than in pipe flanges of similar size: a $\frac{5}{8}$-inch bolt should be the smallest size used for attaching boiler mountings,—even for the smallest valve or cock.

In Naval work the boiler stop-valves are generally made entirely of bronze, and are of the non-return type: non-return valves are also placed at the various bulkheads, in order to localise as far as possible the effects of injury to boilers or pipes by shot, &c.

**Internal steam pipes.**—Where the steam room in a boiler is small relatively to the I.H.P. derived from the boiler, internal pipes with closed ends, and provided with sufficient narrow transverse slits (saw-cuts), or small holes, to give a clear area equal at least to twice that of the pipe section, should be fitted. They are best made of sheet brass —*not* copper. The number and arrangement of the pipes must be determined in accordance with the conditions of the case, but a good plan is to fit two pipes running the full length of the boiler; so that, with the stop-valve at one end, there are two pipes leading to it, each rather more than half the sectional area of the valve or of the branch steam pipe from the boiler, and with the stop-valve on the shell at mid-length there are four shorter pipes converging to it.

The steam is then gently collected from a large area, and strong currents, which might induce priming, are avoided.

**Safety-valves.**—The size of safety-valve should be such that it is capable of discharging all the steam that can be generated in the boiler, without allowing the pressure to rise more than 10 per cent. above that to which the valve is nominally loaded; it therefore depends mainly on quantity of fuel burnt per hour and the working pressure. A convenient and easily applied rule is,—

**Rule 275a. Area in square inches of each of two valves** =

$$\left(\frac{\text{Grate area}}{20} + \frac{\text{Heating surface}}{200}\right) \times \sqrt{\frac{100}{\text{Working pressure}}}.$$

**Rule 276 (Board of Trade). Area of safety-valves in square inches** $=\frac{37\cdot5 \times \text{grate area in sq. ft.}}{\text{absolute pressure in boiler}}$: for natural draught. (For forced draught, see p. 403.)

**Rule 277. B.M.E.D. & C. Committee Rule for safety-valves.**—Aggregate area in square inches = total heating surface in sq. ft. $\times \frac{K}{p+15}$, where $p$ is the working pressure, K is 1·25 for coal-fired boilers, and 1·5 for oil-fired ones.

No valve to have a load exceeding 2600 lbs.

Cross-section of waste steam pipe area in square inches = 0·01 × T.H. surf. in sq. ft.

**Rule 278 (French Govt.). Diameter of valve if one only** $=1\cdot23\sqrt{\frac{\text{T.H.S.}}{p+9}}$. (In English measures.)

**Rule 279 (German Govt.). The clear area through valve seats to be** so many square millimetres per square metre of total heating surface, the range is from 131 for a pressure of 75 lbs. to 51 for 240 lbs.

Wherever possible, the valves should be placed with the spindles vertical, as the action of the valve (which is very sensitive to, and easily affected by, any increase in the friction of its various parts) is then more certain. For the same reason all the parts should be a very easy fit, and the rubbing surfaces of the spindle should be draw-filed and polished with emery cloth in the same direction.

The whole of the parts must be cleaned from time to time, when in use, and the greatest care must be taken to avoid making any dints or burrs on the working surfaces.

The spring should be so proportioned that its initial compression at load is not less than half the diameter of the valve ; this result can be obtained with various proportions of spring, and when height is limited a shorter spring of larger diameter and section of steel may be used.

In Naval practice, the valves, valve-boxes, &c., are usually made entirely of bronze, and the valve is generally made as a separate piece (not cast with the spindle), while the casing gear, when pressure is moderate, may be fitted to lift the valve from below. This construction has the advantage of preventing any bending or springing of the spindle that may arise from the ends of the spring not being quite true and square by its axis, from affecting the tightness of the valve on its seat ; and also permits examination of the valves to be made without disconnecting and taking down easing gear shafts, &c.

The valve faces should always be flat, *not* angled at all.

When the design of the valve permits, it is a great convenience to those who will afterwards have charge of the machinery, to have a thread cut on the upper end of the spindle, so that, before taking the valve to pieces for examination or re-grinding, a nut may be put on to prevent the release or expansion of the spring when the joint at the base of the spring tower is broken ; spring, tower, spindle and valve can then be readily and quickly removed in one piece.

When loose valve sheets are used (as with cast-iron and steel chests) they should be securely fixed in place by a flange, or lugs, with studs or screws.

The valves, seats, spindles, compressing screws and nuts, spring washers, spindle bushes and cotters, studs and nuts for valve seats, and bushes for bearings of easing gear shafts, should be of special bronze suitable for high temperatures.

The valve may be with or without lip, but valves without lip are much less violent in action.

The point of the compressing screw should be well rounded and should enter from $\frac{3}{8}$ to $\frac{3}{4}$ inch into the spring washer.

**Spiral springs.**—The size of steel required by the Board of Trade is given by,—

**Rule 280.**

$$d=\sqrt[3]{\frac{S\times D}{C}},$$

where $d$ = diameter or side of square of steel *in inches* (min. $\frac{1}{4}$ inch).
$D$ = diameter of coil (centre to centre of wire) in inches.
$S$ = load on spring in pounds.
$C$ = 8000 for round and 11,000 for square steel.

In Naval work the values adopted for C are commonly 11,000 and 15,000 for round and square steel respectively.

$d$ should, as recommended by the late T. W. Traill, always equal $\frac{D}{5}$: when this proportion of spring is adopted the above rule becomes,—

$$\sqrt{\frac{S}{1600}}=d\text{ ; or }1600d^2=S\text{ for round steel ;}$$

$$\sqrt{\frac{S}{2200}}=d\text{ ; or }2200d^2=S\text{ for square steel.}$$

The relation between load on, and compression of, spiral springs is given by,—

**Rule 281.**

$$\text{Compression in inches}=\frac{D^3\times S\times N}{d^4\times a},$$

where $N$ = number of free coils (not counting those which are in contact);
$d$ = diameter or side of square of wire in *sixteenths of an inch* ;
$a$ = 26 for round, 32 for square steel ;

and the other symbols have the same meanings as above.

### Table CXVIII.—Safety-Valve Springs by Board of Trade Rules and the above formula.

| Dimensions in inches. | | Working loads in lbs. | | Dimensions in inches. | | Working loads in lbs. | |
|---|---|---|---|---|---|---|---|
| Diameter or side of square of steel. | Mean diameter of coil. | Round steel. | Square steel. | Diameter of side of square of steel. | Mean diameter of coil. | Round steel. | Square steel. |
| $\frac{1}{4}$ | $1\frac{1}{4}$ | 100 | 137 | $\frac{25}{32}$ | $3\frac{29}{32}$ | 976 | 1342 |
| $\frac{9}{32}$ | $1\frac{13}{32}$ | 126 | 174 | $\frac{13}{16}$ | $4\frac{1}{16}$ | 1056 | 1452 |
| $\frac{5}{16}$ | $1\frac{9}{16}$ | 156 | 214 | $\frac{27}{32}$ | $4\frac{7}{32}$ | 1139 | 1566 |
| $\frac{11}{32}$ | $1\frac{23}{32}$ | 189 | 260 | $\frac{7}{8}$ | $4\frac{3}{8}$ | 1225 | 1684 |
| $\frac{3}{8}$ | $1\frac{7}{8}$ | 225 | 309 | $\frac{29}{32}$ | $4\frac{17}{32}$ | 1314 | 1806 |
| $\frac{13}{32}$ | $2\frac{1}{32}$ | 264 | 363 | $\frac{15}{16}$ | $4\frac{11}{16}$ | 1406 | 1933 |
| $\frac{7}{16}$ | $2\frac{3}{16}$ | 306 | 421 | $\frac{31}{32}$ | $4\frac{27}{32}$ | 1501 | 2064 |
| $\frac{15}{32}$ | $2\frac{11}{32}$ | 351 | 483 | 1 | 5 | 1600 | 2200 |
| $\frac{1}{2}$ | $2\frac{1}{2}$ | 400 | 550 | $1\frac{1}{32}$ | $5\frac{5}{32}$ | 1701 | 2339 |
| $\frac{17}{32}$ | $2\frac{21}{32}$ | 451 | 620 | $1\frac{1}{16}$ | $5\frac{5}{16}$ | 1806 | 2483 |
| $\frac{9}{16}$ | $2\frac{13}{16}$ | 506 | 696 | $1\frac{3}{32}$ | $5\frac{15}{32}$ | 1914 | 2631 |
| $\frac{19}{32}$ | $2\frac{31}{32}$ | 564 | 775 | $1\frac{1}{8}$ | $5\frac{5}{8}$ | 2025 | 2784 |
| $\frac{5}{8}$ | $3\frac{1}{8}$ | 625 | 859 | $1\frac{5}{32}$ | $5\frac{25}{32}$ | 2139 | 2941 |
| $\frac{21}{32}$ | $3\frac{9}{32}$ | 689 | 947 | $1\frac{3}{16}$ | $5\frac{15}{16}$ | 2256 | 3102 |
| $\frac{11}{16}$ | $3\frac{7}{16}$ | 756 | 1039 | $1\frac{7}{32}$ | $6\frac{3}{32}$ | 2376 | 3267 |
| $\frac{23}{32}$ | $3\frac{19}{32}$ | 826 | 1136 | $1\frac{1}{4}$ | $6\frac{1}{4}$ | 2500 | 3437 |
| $\frac{3}{4}$ | $3\frac{3}{4}$ | 900 | 1237 | ... | ... | ... | ... |

The late Mr Traill fixed the values of $a$ at 22 and 30 from experiments he had made, whilst Rankine gave 20 to 23 for round and 28·6 to 32·9 for square steel, and Mr Hartnell's experiments (quoted by Unwin) place the values for round steel at 21 to 24·8,—the higher figure for ¼-inch wire, and the lower for ⅜-inch.

The superior transverse elasticity of the round sections, and especially of small ones, compared with square ones, is probably due to the fact that the round wires are often drawn, whilst the square ones are only rolled; also the material of square section steel is probably more affected by the coiling process than is that of the round.

**Feed check valves.**—The sizes of feed pipes may be determined in accordance with the rules laid down in the section on "Feed pumps," and the internal diameter of the valve seat of check valve should be at least ¼ inch more than the diameter of pipe so found.

**Rule 282. Area through main feed valve in square inches** = total heating surface in square feet ÷ 250; or

**Minimum area of pipe section** = water supplied in lbs. per hour ÷ 2500;

$$\text{say, Diameter of main feed pipe} = \frac{\sqrt{\text{pounds water per hour}}}{45};$$

**Rule 282a.**

$$\text{and, Diameter of auxiliary feed} = \frac{\sqrt{\text{pounds water per hour}}}{49}.$$

$$\text{or, Area of section of auxiliary feed} = \text{T.H.S. sq. ft.} \div 300.$$

Feed check valves should be very strongly made, and entirely of the best bronze; the necks by which they are attached to the boilers should be specially strong, and well ribbed to the flanges. The spigots should be turned, and the holes in the boiler plates carefully bored and faced to suit. The working faces of the valves should be flat, and proportioned in accordance with Rule 153 (p. 169). The spindles should be very stout, and provided with square threaded screws; if space allows the nuts to be placed in external crossheads or bridges, they are better so arranged.

The Admiralty require that the main feed check valve shall always be placed at the right hand, as one faces the boiler and the auxiliary valve at the left hand.

**Internal feed pipes.**—The feed water should be led by an internal pipe (of brass) placed 2 or 3 inches below the water level, to a part of the boiler where there is a descending current, and there delivered downwards through a number of fine transverse slits or small holes, Care must be taken that this internal pipe is arranged to be kept always filled with water, and not partly with steam, so as to cause a severe water hammering action to be set up, the joints started, and the pipe very quickly destroyed.

**Blow-off and scum valves.**—These should also be stoutly made valves of bronze,—ribbed as above directed for feed check valves,—and are best fitted so that the pressure tends to hold the valve on its seat, in order to reduce the risk of leakage as much as possible. Now that fresh water is so much used, and loss made up by means of evaporators, and from reserve tanks, the blow-off valve is not so necessary as formerly, and is now often omitted, the boilers being emptied into the bilges when cold.

The scum valve should have an internal pipe leading to it from a circular "scum pan" or dished plate of sheet brass (about 15 inches diameter) fixed near to the centre of the water surface, and at about the lowest working level; or, if preferred, the internal pipe may reach a little beyond the centre of water surface, and may be closed at the end, and have simply a few longitudinal slits in its upper side to admit the scum.

The clear area through the blow-off valve may be that given by,—

**Rule 283.**

$$\text{Area of blow-off pipe (square inches)} = 1 + \frac{\text{Tons of water in boiler}}{5}.$$

The area of scum pipe may be one-third that of blow-off pipe.

**Water-gauge.**—All boilers, axially fore and aft, excepting small donkey boilers, should have two sets of water-gauge fittings, and large double-ended boilers are better with three sets,—one of which may be fitted direct to the shell.

The gauge cocks and glass tubes should be carried on bronze stand-pipes, from which copper pipes should be led to the steam and water spaces of the boilers.

### Table CXIX.—Weight of Pure Water at Different Temperatures.

| Temperature of Water. | Absolute Pressure of Steam. | Weight in Lbs. | |
|---|---|---|---|
| | | A Cub. Ft. | A Gallon. |
| F.° | lbs. | | |
| 60 | 0·2 | 62·37 | 10·0025 |
| 90 | 0·7 | 62·13 | 9·964 |
| 120 | 1·7 | 61·74 | 9·901 |
| 136 | 2·7 | 61·45 | 9·858 |
| 150 | 3·7 | 61·18 | 9·812 |
| 160 | 4·7 | 60·98 | 9·780 |
| 168 | 5·7 | 60·81 | 9·752 |
| 175 | 6·7 | 60·66 | 9·728 |
| 181 | 7·7 | 60·53 | 9·707 |
| 187 | 8·7 | 60·39 | 9·685 |
| 192 | 9·7 | 60·27 | 9·666 |
| 196 | 10·7 | 60·17 | 9·650 |
| 201 | 11·7 | 60·05 | 9·630 |
| 205 | 12·7 | 59·95 | 9·614 |
| 209 | 13·7 | 59·84 | 9·596 |
| 212 | 14·7 | 59·76 | 9·584 |
| 215 | 15·7 | 59·74 | 9·581 |
| 228 | 20 | 59·43 | 9·532 |
| 240 | 25 | 59·10 | 9·479 |
| 250 | 30 | 58·75 | 9·430 |
| 259 | 35 | 58·54 | 9·389 |
| 267 | 40 | 58·30 | 9·349 |
| 275 | 45 | 58·04 | 9·312 |
| 281 | 50 | 57·86 | 9·300 |
| 293 | 60 | 57·50 | 9·220 |
| 303 | 70 | 57·16 | 9·166 |
| 312 | 80 | 56·85 | 9·117 |
| 320 | 90 | 56·57 | 9·072 |
| 327 | 100 | 56·33 | 9·035 |
| 334 | 110 | 56·07 | 8·991 |
| 341 | 120 | 55·83 | 8·953 |
| 347 | 130 | 55·62 | 8·920 |
| 353 | 140 | 55·41 | 8·885 |
| 358 | 150 | 55·21 | 8·855 |
| 363 | 160 | 55·02 | 8·824 |
| 368 | 170 | 54·84 | 8·795 |
| 373 | 180 | 54·67 | 8·766 |
| 377 | 190 | 54·50 | 8·741 |
| 381 | 200 | 54·34 | 8·717 |
| 385 | 210 | 54·20 | 8·693 |
| 389 | 220 | 54·03 | 8·668 |
| 393 | 230 | 53·87 | 8·650 |
| 397 | 240 | 53·75 | 8·619 |
| 400 | 250 | 53·63 | 8·590 |
| 404 | 260 | 53·46 | 8·574 |
| 407 | 270 | 53·39 | 8·558 |
| 410 | 280 | 53·34 | 8·548 |
| 413 | 290 | 53·32 | 8·540 |
| 416 | 300 | 53·31 | 8·538 |

The internal diameter of the stand-pipe and its connections should not be less than 1 inch, and for large boilers 1½ inches is desirable, and the connections should be made of extra stout piping,—say, not thinner than 9 L.S.G. for 160 lbs. pressure and upwards.

If cocks or valves are to be fitted where these connections join the boiler, they should be in sight and not capable of being closed without advertising the fact,—as serious accidents have occurred through their use,—and care should be taken not to place the inlets near where a current of steam or water may be, since the level of the water in the glass may be affected thereby,—a difference of pressure of one-tenth of a pound causing an alteration in level of 2·7 inches.

Also, in fixing the level at which the water is to stand in the glass, the lowering of level due to the cooling and consequent contraction of the water in the stand-pipe connection should not be forgotten,—though it is always an error on the side of safety.

Rankine gave the following rule for determining the volume of water at different temperatures:—

Let the volume of the water at its temperature of maximum density (39·2° F.) be represented by unity, and let its volume at T° be V, then,—

**Rule 284.**
$$V=\frac{1}{2}\left(\frac{T+461}{500}+\frac{500}{T+461}\right).$$

He stated that the error is only one four-hundredth.

**Circulating apparatus.**—It is very desirable that large rigidly constructed boilers should be fitted with an apparatus for circulating the water whilst getting up steam; if hydrokineters are not supplied, connections should be made with one of the auxiliary steam pumps.

In almost all cylindrical boilers the rate of evaporation may be improved by fitting a proper arrangement of circulating plates; the first cost is small, but they are rather in the way when the boiler is being cleaned; the gain, however, is often worth the cost and trouble in crowded boilers which have to be forced.

**An air-cock** should be fitted at the highest point of each boiler.

## THE BOARD OF TRADE RULES FOR BOILER MOUNTINGS, &c.

185. No boiler or steam chamber should be so constructed, fitted, or arranged that the escape of steam from it through the safety-valves required by the Act of Parliament can be wholly, or partially intercepted by the action of another valve.

A stop-valve should always be fitted between the boiler and the steam pipe, and, where two or more boilers are connected with a steam receiver or superheated, between each boiler and the superheater or steam receiver.* The necks of stop-valves should be as short as practicable and the chests should be tested when new to double the water pressure.

186. **Water-gauges, test cocks, and gauges.**—Each boiler should be fitted with a glass water-gauge, at least three test cocks, and a steam gauge. Boilers that are fired from both ends, and those of unusual width, should have a glass water-gauge and three test cocks at each end

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* The object of this is obvious, viz. to avoid the failure of all the boilers through the failure of one.

or side, as the case may be. An additional glass water-gauge may, however, be substituted for three test cocks. When a steamer has more than one boiler, each boiler should be treated as a separate one, and have all the requisite fittings.

When the water-gauge cocks are not attached directly to the shell of the boiler, but to a stand-pipe or column, cocks should as a general rule be fitted between the boiler and the stand-pipes, &c., and may be placed either on the boiler or at the stand-pipe. Such cocks need not, however, be insisted on in cases where the columns, stand-pipes, &c., are of moderate length and of suitable strength, provided that the diameter of the bore of any part is not less than 3 inches.

*Valves* between boiler and stand-pipe should not be passed.

If the column, stand-pipes, &c., are of less diameter than 3 inches, and the pipes are bolted to the boiler without the intervention of cocks, the arrangement need not be objected to, if otherwise satisfactory, providing there is no difficulty in keeping the passages at the ends clear, and ascertaining that they are so. To do this it will be necessary that the passage in the part of the column between the top and bottom gauge-glass cocks be cut off or closed, which may be done permanently, or by the interposition of a cock at that part. The latter is a convenient and desirable arrangement even when cocks are fitted on the boiler.

In the case of high-pressure boilers, it is desirable that the cocks in connection with the water-gauges should be fitted with handles which can be expeditiously manipulated from a convenient position.

It is desirable in all cases that test cocks should be fitted directly to the skin of the boiler; and when the water-gauge is attached to a column, the opening through which is stopped or can be cut off, the test cocks *must* be fitted directly to the skin of the boiler.

The Surveyors should satisfy themselves by actual examination whether the glass water-gauges of the boilers of the vessels they survey are clear, and also whether they are fitted with automatic valves or fittings, as the existence of such fittings cannot always be ascertained by external examination. In all cases where automatic gauges are fitted, full particulars thereof should be submitted for consideration and approval before the gauges are passed.

191. Cast-iron stand-pipes or cocks intended for the passage through them of hot brine should not be passed.

Surveyors should also discourage the use of cast-iron chocks and saddles for boilers, and particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

### The Board of Trade Rules for Safety-Valves are as follows :—

177. Cases have come under the notice of the Board of Trade in which there were pipes between the boilers and the safety-valves. Such arrangement is not in accordance with the Act, which distinctly provides that the safety-valves shall be upon the boilers.

The Surveyors are instructed that in all *new boilers*, and whenever *alterations can be easily made*, the valve chest should be placed directly on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible, and be cast in one with the chest.

The Surveyors should note that it is not intended by this instruction that vessels with old boilers which have been previously passed with such an arrangement should be detained for the alterations to be carried out.

Of course, in any case in which a Surveyor is of opinion that it is positively dangerous to have a length of pipe between the boilers and the safety-valve chest, it is his duty at once to insist on the requisite alterations being made before granting a declaration.

If any person place an undue weight on the safety-valve of *any* steamship, or, in the case of steamships surveyed under the Act, increase such weight beyond the limits fixed by the Engineer Surveyor, he shall, in addition to any other liability he may incur by so doing, be liable for each offence to a fine not exceeding one hundred pounds.

179. The locked-up valves, *i.e.* those out of the control of the engineer when steam is up, should have an area not less, and a pressure not greater, than those which are not locked up, if any such valves are fitted.

When natural draught is used, the area per square foot of fire-grate surface of the locked-up safety-valves should not be less than that given in the following Table opposite the boiler pressure intended, but in no case should the valves be less than 2 inches in diameter. This applies to new vessels or vessels which have not received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the Table, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits.

When forced draught is used, the area of the safety-valves should not be less than that found by the following formula :—

$$\frac{A \times C}{20} = \text{area of valves required,}$$

where A = area of valves as found from the Table,
C = estimated coal consumption in lbs. per square foot of grate per hour.

When the pressure exceeds 180 lbs. per square inch the accumulation of pressure at the steam test will probably be exceptionally high, unless the area of the branch leading from the valve chest is in excess of the area of the valves, and the area of the main waste steam pipe is correspondingly in excess of the gross area of the valves.

In ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

In the case of vessels that have not had a passenger certificate, if there is only one safety-valve on any boiler, the Surveyor should not grant a declaration without first referring the case to the Board for special instructions.

## Table CXX.—Safety-Valve Areas (Board of Trade) (Natural Draught).

| Boiler Pressure. | Area of Valve per square foot of Fire-grate. | Boiler Pressure. | Area of Valve per square foot of Fire-grate. | Boiler Pressure. | Area of Valve per square foot of Fire-grate. | Boiler Pressure. | Area of Valve per square foot of Fire-grate. |
|---|---|---|---|---|---|---|---|
| 132 | ·255 | 154 | ·221 | 176 | ·196 | 198 | ·176 |
| 133 | ·253 | 155 | ·220 | 177 | ·195 | 199 | ·175 |
| 134 | ·251 | 156 | ·219 | 178 | ·194 | 200 | ·174 |
| 135 | ·250 | 157 | ·218 | 179 | ·193 | 201 | ·173 |
| 136 | ·248 | 158 | ·216 | 180 | ·192 | 202 | ·173 |
| 137 | ·246 | 159 | ·215 | 181 | ·191 | 203 | ·172 |
| 138 | ·245 | 160 | ·214 | 182 | ·190 | 204 | ·171 |
| 139 | ·243 | 161 | ·213 | 183 | ·189 | 205 | ·170 |
| 140 | ·241 | 162 | ·211 | 184 | ·188 | 206 | ·169 |
| 141 | ·240 | 163 | ·210 | 185 | ·187 | 207 | ·169 |
| 142 | ·238 | 164 | ·209 | 186 | ·186 | 208 | ·168 |
| 143 | ·237 | 165 | ·208 | 187 | ·185 | 209 | ·167 |
| 144 | ·235 | 166 | ·207 | 188 | ·184 | 210 | ·166 |
| 145 | ·234 | 167 | ·206 | 189 | ·183 | 211 | ·166 |
| 146 | ·232 | 168 | ·204 | 190 | ·182 | 212 | ·165 |
| 147 | ·231 | 169 | ·203 | 191 | ·181 | 213 | ·164 |
| 148 | ·230 | 170 | ·202 | 192 | ·181 | 214 | ·164 |
| 149 | ·228 | 171 | ·201 | 193 | ·180 | 215 | ·163 |
| 150 | ·227 | 172 | ·200 | 194 | ·179 | 216 | ·162 |
| 151 | ·225 | 173 | ·199 | 195 | ·178 | 217 | ·161 |
| 152 | ·224 | 174 | ·198 | 196 | ·177 | 218 | ·161 |
| 153 | ·223 | 175 | ·197 | 197 | ·176 | 219 | ·160 |
|  |  |  |  |  |  | 220 | ·159 |

180. The safety-valves should be fitted with lifting-gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting-gear should in all cases be so arranged that it can be worked b hand either from the engine-room or stoke-hold.

Care should be taken that the safety-valves have a lift equal to one-fourth of their diameter; that the openings for the passage of steam to and from the valves, including the waste-steam pipe, have each an area not smaller than the area required by Section 179, and the area of the main waste-steam pipe should not be smaller than the combined area of the branch pipes. Each valve box should have a drain pipe fitted at its lowest part.

Too much care cannot be devoted to seeing that there is proper lift, also that free means of escape for the waste steam are provided, as it is obvious that, unless the means for escape of the waste steam are ample, the effect is the same as reducing the area of the valves or putting an extra load upon them. The valve seats should be secured by studs and nuts.

181. The following conditions are to be complied with :—

(1) That at least two valves are fitted to each boiler.
(2) That the valves are of the proper size, as by clause 179.
(3) That the springs and valves are so cased in that they cannot be tampered with.
(4) That provision is made to prevent the valves flying off in case of the springs breaking.
(5) That screw lifting-gear is provided to ease all the valves, as by clause 180.
(6) That the size of the steel of which the springs are made is in accordance with that found by the following formula :—

$$\sqrt[3]{\frac{s \times D}{c}} = d :$$

$s$ = the load on the spring in lbs.
D = the diameter of the spring (from centre to centre of wire) in inches.
$d$ = the diameter, or side of square, of the wire in inches.
$c$ = 8000 for round steel.
$c$ = 11,000 for square steel.

(7) That the springs are protected from the steam and impurities issuing from the valves.
(8) That when valves are loaded by direct springs, the compressing screws abut against metal stops or washers, when the loads sanctioned by the Surveyor are on the valves.
(9) That the springs have a sufficient number of coils to allow a compression under the working load of at least one quarter the diameter of the valve.

The size of steel of springs of safety-valves should not as a rule be less than $\frac{1}{4}$ inch.

182. **Safety-valve steam tests.**—In no case is the Surveyor to give a declaration for spring-loaded valves, unless he has tried them under full steam, and full firing, for at least 15 minutes with the feed-water shut off and stop-valve closed, and is fully satisfied with the result of the test. If the accumulation of pressure exceed 10 per cent. of the loaded pressure he should withhold his declaration and report the case to the Board of Trade.

183. The tracings of new safety-valve designs should, if possible, be transmitted to the Board of Trade for consideration before the construction of the safety-valves is commenced.

178. **Liability of owners, &c.**—It is clearly the duty of the

masters and engineers of vessels to see, in the intervals between the surveys, that the locked-up safety-valves, as well as the other safety-valves, and the rest of the machinery, are in proper working order. There is no provision in the Merchant Shipping Act, 1894, exempting the owner of any vessel, on the ground that she has been surveyed by the Board of Trade Surveyors, from any liability, civil or criminal, to which he would otherwise be subject. The Act of Parliament requires the Government safety-valves to be out of the control of the engineer when the steam is up; this enactment, far from implying that he is not to have access to them, and to see to their working, at proper intervals when the vessel is in port, rather implies the contrary; and the master should take care that the engineer has access to them for that purpose. Substantial locks that cannot be easily tampered with, and as far as possible weather-proof, should be used for locking up the safety-valve boxes.

In witnessing the tests of boilers, &c., and safety-valves, the Surveyors are to use the pressure gauges supplied by the Board of Trade. The steam gauge should not be used without a syphon filled with water between it and the boiler.

## B.M.E.D. & C. COMMITTEE RULES FOR SAFETY-VALVES.

At least two safety-valves must be fitted to each boiler. They must be arranged so that the springs and valves are cased in, that the valves cannot be overloaded when steam is up, that they can be lifted by easing-gear, and turned round on their seats by hand, and in case of fracture of springs they cannot lift out of their seats.

**The easing-gear** must be arranged to lift all the safety-valves on a boiler together, and must be workable from some accessible place free from steam danger. All joints must have bronze bushes.

**Maximum size of safety-valves.**—No safety-valve should have a load on it exceeding 2600 lbs. If larger valve area is required than two valves give, it must be provided by increasing the number of valves.

**All the safety-valves** of each boiler may be fitted in one chest, which must be connected direct to the boiler by a strong and stiff neck, the passage through which should have a cross-sectional area equal to one-half the aggregate area of the valves in the chest.

**Size of safety-valves** of ordinary tank boilers—

$$\text{Aggregate area of valves in square inches} = \text{total heating surface in square feet} \times \frac{K}{p+15},$$

$p$ is the working pressure in lbs. per square inch, K for coal-fired boilers is 1·25 and for oil-fired 1·5.

**The waste-steam pipe** = total heating surface in square feet ÷ 100 = area of cross-section in square inches.

## BOILER MOUNTINGS AND FITTINGS, B.M.E.D. & C. COMMITTEE.

**Water gauges**, two to each boiler. Tubes ¾ inch or ⅝ inch outside diameter, and 20, 18, 16, or 14 inches long. Double-ended boilers over 16 feet diameter to have three. Stand pipes at least 2½ inches diameter inside.

**Pressure gauge**, one to single-ended, two to double-ended boilers.

**Salinometer cock**, one fitted direct to each boiler.

**Check feed valves**, two independent valves to each boiler, with shut-off cocks or valves between them and the boiler.

**Blow-off valve** to each boiler not more than 1½ inches diameter.

**Stop-valves for steam**, one main, one auxiliary to each.

**Valves for steam steering and whistles**; and when more than one boiler, two at least shall have these valves.

**All valves over** 1½ inches diameter must have outside screws, and shut with right-hand motion.

**All cocks and valves** shall be such as to be seen whether open or shut.

**Stop and safety valves**, chests for temperatures over 425° F. must be of cast steel.

**Feed valves, chests**, and all cocks and valves should be of strong tough bronze, and those exposed to steam should be such as to maintain good strength and toughness while at the steam temperatures.

All chests of stop-valves, safety-valves, check-feed valves, and the shells of cocks over one inch diameter shall be tested to twice the working pressure.

## LLOYD'S RULES FOR BOILER MOUNTINGS, &c.

5. Two safety-valves are to be fitted to each boiler, and loaded to the working pressure in the presence of a Surveyor. In the case of boilers of greater working pressure than 60 lbs. per square inch, the safety-valves may be loaded to 5 lbs. above the working pressure. If common valves are used, their combined areas are to be at least half a square inch to each square foot of grate surface. If improved valves are used, they are to be tested under steam in the presence of the Surveyor; the accumulation is in no case to exceed 10 per cent. of the working pressure.

6. An approved safety-valve is also to be fitted to the super-heater.

7. In winch boilers one safety-valve will be allowed, provided its area is not less than half a square inch per square foot of grate surface.

8. Each valve is to be arranged so that no extra load can be added when steam is up, and must be fitted with easing gear which must lift the valve itself. All safety-valve spindles are to extend through the covers, and are to be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

9. Stop valves are to be fitted so that each boiler can be worked separately.

10. Each boiler is to be fitted with a separate steam gauge, to accurately indicate the pressure.

11. Each boiler is to be fitted with a blow-off cock, independent of that on the vessel's outside plating.

## FURNACE FITTINGS.

**Furnace fronts, &c.**—Furnace fronts and fire doors may with advantage be of wrought-iron or steel,—although they are often of cast-iron; but the internal protecting or baffle plates are better of cast-iron, since it burns away less rapidly than the wrought material, and should be in several small pieces, free to expand in all directions, rather than in one large piece, which would probably very soon crack and get adrift.

The size of fire door, in the clear, may vary from 12 inches high by 18 inches wide, to 16 inches high by 24 inches wide,—the top of the opening being arched, and struck with a radius equal to the *width* of the opening.

A very good arrangement is that in which the otherwise useless corners to right and left of, and above the fire door, inside the furnace, are filled in by curved cast-iron plates, perforated with small holes to allow the air to enter the furnace; the air is thus heated to a certain extent in these boxes or chambers before coming into contact with the fuel, and the riveted joint round the furnace mouth is protected from the fire.

Furnaces over 3 feet 6 inches diameter are perhaps better with two half-doors hinged right and left, and meeting in the centre, only one of which need be opened at once; less cold air is then admitted, and one side of the fire can be attended to at a time.

Means should be provided for holding the doors open in a sea-way.

**Fire-bars, &c.**—When chimney draught only, or chimney and forced draught not exceeding ·5 inches of water, is used, the bars are better in one length,—up to 5 feet 6 inches long; when a greater air pressure is used, or when the grate is longer than 5 feet 6 inches, two lengths of bars may be used.

When one long bar is used, it should be hooked to the inner edge of the dead plate, and free to expand inwards, and slide on the bridge-plate; when two lengths are used, they should be hooked to the central bearer-bar, and free to slide on both the bridge-plate and the dead-plate.

In either case care should be taken that the dead-plate is formed so as to hold or support the bars with their faces or upper surfaces flush with its own upper surface.

The thickness of bar and width of air space between bars must depend on the class of coal that will generally be used, and on the air pressure with which it is intended to work.

The following are good average dimensions for fire-bars:—

| | |
|---|---|
| Thickness on face, . . . . . . . | 13/16 inch. |
| ,, at bottom edge, . . . . | ⅜ ,, |
| ,, on face, over distance pieces, . . | 1 3/16 ,, |
| ,, at bottom edge, ,, ,, . . | 1⅛ ,, |
| Depth at centre, . . . . | $\cdot 66\sqrt{\text{length}}$ in inches. |
| ,, near ends, . . . . . . | 2½ inch. |
| Width of air spaces, . . . . . . | ⅜ ,, |

The slope of the grate surface should never be less than 1 inch per foot of length, and is better 1½ inches, or even more, as is possible with large ones and short bars.

When corrugated or ribbed furnaces are used, the two side bars should be made to templates from the furnaces, and should fit as closely as possible into all the recesses.

Fire-bars should, of course, be made of the most refractory iron obtainable; fine grey irons are quite unsuitable.

**Bridges, &c.**—In return-tube boilers the grate should never be so long that the front face of the bridge is less than 9 inches from the face of the back tube-plate.

The height of bridge should be such that the clear area above it may be from ⅙th to ⅛th of the area of grate; this proportion is obtained approximately when the clear height above the bridge at its centre is ⅓rd of the diameter of furnace.

In the case of corrugated or grooved furnaces, the ash-pits require to be fitted with thin lining plates to enable the rake to be used. In Naval and other ships in which forced draught is used to any considerable extent, shallow pans which can be kept full of water are necessary.

Each ash-pit should also be fitted with a good stout pricker bar and with a damper; when the closed stoke-hole system of forced draught is used, the dampers are sometimes balanced and made to open inwards only,—closing against any pressure that may come from the furnace side.

## LADDERS AND PLATFORMS, &c.

The following Table gives ordinary dimensions of ladders and gratings:—

### Table CXXI.—Ladders and Gratings.

| Ladders. | | | | | "Spill" Gratings. | | | |
|---|---|---|---|---|---|---|---|---|
| Width of ladder. | Size of side bars. | | Width of cast-iron steps. | Bar steps, No., size, and pitch of bars. | Width of grating. | Size of side bars. | Diameter of "spills." | Pitch of "spills." |
| | Cast-iron steps. | Round bar-iron steps. | | | | | | |
| in. | in. in. | in. in. | in. | in. | in. in. | in. in. | in. | in. |
| 12 | ... | $2\frac{1}{4} \times \frac{3}{8}$ | ... | one $\frac{5}{8}$ | 15 & 18 | $2\frac{3}{4} \times \frac{3}{8}$ | $\frac{5}{8}$ | $2\frac{1}{4}$ |
| 15 | ... | $3\frac{1}{4} \times \frac{3}{8}$ | ... | two $\frac{5}{8}$<br>$1\frac{3}{8}$ pitch | 21 | $3 \times \frac{7}{16}$ | $\frac{3}{4}$ | $2\frac{3}{8}$ |
| 18 | $3\frac{1}{2} \times \frac{3}{8}$ | $3\frac{1}{2} \times \frac{3}{8}$ | $4\frac{1}{4}$ | two $\frac{5}{8}$<br>$1\frac{1}{2}$ pitch | 24 | $3 \times \frac{7}{16}$ | $\frac{3}{4}$ | $2\frac{3}{8}$ |
| 21 | $3\frac{3}{4} \times \frac{7}{16}$ | ... | $4\frac{3}{4}$ | ... | 27 | $3 \times \frac{7}{16}$ | $\frac{3}{4}$ | $2\frac{3}{8}$ |
| 24 | $4 \times \frac{7}{16}$ | ... | $5\frac{1}{4}$ | ... | 30 and above | $3 \times \frac{7}{16}$ | $\frac{5}{8}$ with centre support | $2\frac{1}{4}$ |

⅝-inch round iron "spills" should not be used for spans over 18 inches.

¾-inch round iron "spills" should not be used for spans over 27 inches.

$d$ = Diameter of spills in sixteenths.
L = Length in inches.

Then
$$L = \frac{d^3}{70} + 4.$$

Ladder steps should be from 9 to 10 inches apart (face to face).

The front corners of cast-iron steps should be rounded away, and they should be attached to the side bars of the ladder by two ½-inch bolts at each end.

The main engine-room ladder should be at least 21 inches wide, and in large ships, where there is plenty of room, a width of 24 to 30 inches.

Stoke-hole ladders are not usually fitted with cast-iron steps.

The inclination of a ladder to the vertical may be almost anything, and depends on the space available and purpose for which ladder is fitted; the main engine-room ladder should be 1 in 2¼ when possible.

Handrails should be of steel or wrought-iron, 1 inch in diameter.

Stanchions, when 3 feet high, may taper from ⅞ inch diameter at the top to 1¼ inches at the bottom; when short, for ladders, ¾ inch at top to 1 inch at bottom is enough. The ball through which the rail passes may be 2 inches in diameter.

Engine and boiler-room floors are best laid with chequered wrought-iron or steel plates,—engine-room 5/16 inch thick, and boiler-room ⅜ inch, exclusive of the raised pattern. In the Navy, ¼ inch and 5/16 inch respectively are the usual thicknesses.

It is very desirable that floor plates should be secured to the bearers where possible, as an accumulation of water surging from side to side in the stoke-hole may lift the plates and drive them against the sea-valves and pipes, and thus cause serious damage, sufficient sometimes to imperil the safety of the ship.

## ENGINE AND BOILER SEATINGS, &c.

It is extremely difficult to lay down any general rules that will be of service in designing engine and boiler seatings, because so much depends on the type and structure of ship, strength and the design of engine framing or bed-plate, type of boilers and position in which they are to be placed, &c., but the following hints may be of some use.

**Seatings for vertical engines.**—When the seating must be built upon the top edges of the ship's floors, it generally consists of two box girders, one under each side of the bed-plate, parallel to the shaft axis. Whether additional cross girders should be fitted under each main bearing depends on the strength of the framing or bed-plate.

Such girders should be made generally of scantling corresponding with the frames and keelsons of the ship. Their tops should be much thicker, and are usually from 1½ to 2 times the thickness of the vertical plates.

One of the chief difficulties with this type of structure is to get a sufficiently good attachment to the floors; double reverse bars should always be fitted under the engines and boilers, and for large and heavy engines the attachment of the vertical side plates to the reverse bars should also be by means of double angle bars, so that there may be four rivets at every crossing point. The second bar is sometimes represented by a separate short piece at each frame. In very lightly built ships it is often advisable to carry the vertical plates down between the floors, and to attach them to the skin plating as well as to the reverse bars.

Most ships nowadays have double bottoms fore and aft, and on them the engine girders are erected and riveted to them, thereby getting a strong attachment and a good distribution of strains. Cross girders or brackets are usually fitted for this purpose, as also to give stability to the structure and prevent racking.

In all cases where rigid girders are added to the structure of the ship for the purpose of properly distributing weights or strains, care should be taken that they do not stop abruptly at any point, bulkhead or otherwise, but gradually decrease in section, or taper down for three or four frame spaces, as otherwise serious results may ensue from the localisation of the flexure or spring of the structure.

Very large and heavy engines are generally so constructed as to require only a plain flat surface of the same length and breadth as the bed-plate for a seating, and where the ship is built with open floors this should be obtained by placing a number of longitudinal girders across the tops of the floors and plating them over, using only so many athwartship girders as may be absolutely necessary.

In either of the cases, if the shipbuilders are communicated with in time, there will not generally be any difficulty in modifying the spacing of the longitudinals slightly to suit the engines ; a few inches in height may also sometimes be gained by forming troughs or recesses in the inner bottom under the cranks.

Where box girders are placed upon the inner skin as described above, the Admiralty rule is that they shall be made watertight, in order to prevent internal corrosion.

**The seatings for paddle engines** are usually constructed on one of the above described plans, or on a combination of them, and do not require any special description. The cases in which the framing of the engine is constructed of plates and angles and built into the ship are of too special a nature to be advantageously treated here ; and the various methods of stiffening the sides of the ship and attaching the brackets for carrying the outer bearings also fall into the same category.

**Thrust block seatings** should have specially strong and well-extended attachments ; they should extend over at least three frame spaces in small vessels, and over six or more in large ones, and some of the longitudinals of the engine seating should be made continuous with those of the thrust seat.

**The rivet holes** for the thrust blocks and all parts of the engine beds subject to racking should be drilled, or, say, a half of them drilled and the remainder carefully reamered out, otherwise the rivets will work slack and be inefficient.

**Holding down bolts** should be numerous and well distributed, and should have the ends slightly burred over to prevent the slacking back of the nuts.

**Staying of engines.**—The cylinders, &c., of vertical engines may be stayed, but under no circumstances rigidly to the decks or upper works of the ship ; nor should the two sets of engines in a twin-screw vessel be stayed to one another ; there is usually no difficulty in obtaining all necessary stiffness and stability by care in designing the framing, and the risks run by using rigid stays to the ship are sometimes serious. In paddle vessels it is not practically possible to avoid connection between the engines and the upper works of the ship, but even here, where a certain amount of spring is allowed for, cracked entablatures, or top frames, were by no means uncommon.

**Boiler seatings.**—When either single or double-ended boilers are placed with their axes athwartships, the best type of seating is that in which H section girders (10 or 12 inches deep) running the full length of the boiler-room are used, the lower flanges being riveted to double reverse bars, or the inner skin, and the upper flanges carry the wedge-shaped chocks which maintain the boilers in place. When the vessel is of cellular construction these girders should be arranged when possible to coincide with the longitudinals in the bottom.

When the boilers are placed with their axes fore and aft, as is now usual, each cradle or bearer is built upon the top of a separate floor; the proper distribution of the weight is more difficult, but can be effected by putting in additional intercostal longitudinals, or, preferably by fitting longitudinals on top of the floors. Care must be taken that these additional longitudinals do not prevent access to the underside of the boiler; if they are made of considerable depth, manholes may be cut in them. When the construction of the vessel is cellular, there is not usually any need to supplement the longitudinal connections.

Single-ended boilers should have two cradles or pairs of chocks; double-ended boilers of moderate size and weight, three; and very large and heavy double-ended boilers, four.

The boilers should be prevented from moving end-ways by "toe" plates or brackets riveted to some convenient portion of the seating or of the vessel's structure.

The area of section of such toe plates should be not less than $\frac{W \times K}{96}$ square inches to resist shear. W being the weight of the boiler, &c., in tons, and K the speed of the ship in knots.

The greatest care must also be taken so to secure the boilers in their seats that no possible movement of the vessel will throw any strain upon any of the pipes or connections. This is best done by riveting to the upper part of the shell of each boiler four "eye" plates, from each a rod or link is fitted leading to some convenient part of the vessel's structure (such as stringer or deck-beam), or to the neighbouring boiler. These eyes may be riveted in places when the riveting of the shell is done, and they then serve to sling the boiler by. Brackets of plate and angle-iron attached to the deck-beams and almost touching the top of the shell, acting as inverted toe plates, are also sometimes a convenient means of fixing the boilers in their seats.

The common Admiralty method of securing cylindrical boilers was to rivet four eyes to the shell at about a foot above the top of the seatings and in the same planes with them, and four other similar eyes to the tops of the seatings themselves, and then connect each pair of eyes by a pair of flat links and pins, &c.

# LLOYD'S RULES FOR ENGINE AND BOILER ROOMS.

**Section 30.—1. Engine and Boiler Bearers.**—In steam vessels care must be taken that the engine and boiler bearers are properly constructed, and fitted with efficient longitudinal ties. Where the bearers interfere with the longitudinal strength of the vessel, they must extend a sufficient distance beyond the bulkheads of the engine and boiler space to compensate for the same.

**When Engines of High Power are fitted.**—Where it is intended to fit engines of greater power than in ordinary cargo-carrying steamers, the engine seating is to be of proportionately greater strength, and to be specially adapted with this object in view by being connected to the sides of the vessel. Other means are, if necessary, to be adopted in order to ensure the rigidity and strength necessary to withstand the vibration produced in this part of the vessel.

2. **Strengthening in Machinery Space.**—Additional strengthening by means of web frames, and "strong" beams, or otherwise is to be provided in the machinery space. Plans showing the proposed additional strengthening are to be submitted for approval.

3. **Clearance between Bulkheads and Boilers.**—Coal bunker bulkheads are to be kept well clear of the boilers and their uptakes. Where the boiler-room bulkhead is recessed for a donkey boiler, the recess is to be of a size sufficient to give space all round the boiler to admit of its being properly attended to.

In order to afford protection against the heat from the boiler, the roof of the recess is to be not less than 4 feet clear of the top of the boiler, the space between the bunker or hold bulkhead plating and the chimney is to be not less than 18 inches, and a baffle plate is to be fixed between the chimney and the bulkhead; other efficient means may be provided. Wood lining is to be fitted on the hold side of the recess plating with an air space between it and the plating.

4. **Tie-beams across Recess.**—When a recess extending above the hold beams is formed in the engine-room bulkheads, the bulkhead is to be efficiently connected from side to side by a tie or bridle beam at about the height of the hold beams, strongly riveted to the plating and fitted with efficient gusset plates.

5. **Protection of Deck under Donkey Boilers.**—Where vertical donkey boilers are placed on the decks of vessels, the deck underneath them is to be protected by being covered with firebrick or cement not less than 2 inches in thickness. The deck on which fires may be drawn from any donkey boiler is also to be protected by firebrick or cement not less than 2 inches in thickness.

6. **Shaft Tunnel.**—The plating of shaft tunnels is to be of the thickness required in Table 10 for the lower part of bulkhead plating; the top plating in way of the hatchways to be not less than ·10 of an inch thicker than the remaining plates, or to be covered with wood not less than 2 inches thick.

The tunnel is to be strengthened with transverse angle bars of the size of the lower deck stringer angles spaced not more than two frame spaces apart, and 3 feet in way of the hatchways.

The plating is to be caulked, and the tunnel tested with water from a hose to ensure its being watertight.

The bulkheads and top plating of tunnel recesses to be strengthened and supported by similar angles, but spaced the same as the vessel's frames; the top plating where attached to the sides of the vessel to be made watertight with steel or iron collars or chocks, to the exclusion of wood or cement.

The tunnel to be fitted with a watertight sluice door on the engine-room bulkhead, capable of being closed from the upper deck.

If a pipe tunnel is led through the forward holds, its structure is to be the same as that of a shaft tunnel.

**Section 50 (Valves, etc.).**—1. No sluice valve or cock is to be fitted to the collision bulkhead.

2. No sluice valves or cocks are to be fitted to the engine-room or other watertight bulkheads unless they are arranged so as to be accessible at all times.

3. If the after peak is used as a ballast tank, no sluice valve or cock is to be fitted to the after bulkhead; but if it is not so used, and if no pump is fitted in it, a sluice valve or cock is to be fitted to the after bulkhead, to allow water to reach the pumps when required.

4. When sluice valves are fitted, they are to be so arranged as to be controlled from above the load water-line, and the rods are to be boxed in to prevent injury.

5. All head and stern pumps to be provided with sea cocks fitted to the outside plating to the satisfaction of the Surveyors, and in places where they are at all times accessible.

6. Where soil pipes are attached to the outside plating below the load water-line, the lower length must be of steel or iron of substantial thickness, and be secured to the plating with a proper faced joint, and extended for some distance above the load water-line.

7. If the remainder of the pipe be of lead, care must be taken that it is of substantial thickness, and that it is properly protected externally with zinc or iron, to the satisfaction of the Society's Surveyors.

**Section 31 (Openings in Decks).**—1. The engine and boiler openings of the weather-deck of steam vessels are to be properly framed for a height of not less than 18 inches above the deck, the coaming plates to extend to the lower edge of the beams, and iron or steel trunk bulkheads connected to the coamings should be fitted to a height of about 7 feet above the deck.

2. The engine and boiler openings in the 'tween decks of all vessels are also to be enclosed by trunk casings efficiently stiffened by angle bars 30 inches apart, and extending to the weather-deck beams, to which they are secured.

3. Strong iron doors will be allowed in these trunk casings, provided their lower parts are at least 18 inches above the deck, and efficient arrangements made for their security.

4. When a poop, or bridge-house, covers the engine and boiler space, the coamings of the engine and boiler openings should not be less than 2 feet above such deck, unless these openings are constructed as provided for in the first paragraph of this section.

5. It is considered that in all cases the engine and boiler openings should be made as small as practicable, and be subdivided by athwartship iron divisional bulkheads to secure the maximum safety of the vessel. The two sides of the casing should in all instances be efficiently connected by angle beams within them at the upper part.

6. The engine-room skylights are to be in all cases substantially constructed and to be securely bolted or riveted to the coamings, and where the skylight top is not solid with bull's-eyes fitted in the same, efficient dead-lights of metal or wood must be provided. The grating openings over the stoke-hold must also be protected by plates, fitted with hinges, or otherwise in a manner satisfactory to the Surveyors.

7. Where either of the openings exceeds 15 feet, or the combined length exceeds 30 feet, the beams in way of the same are to be plated over from the stringer to the tie-plates, the plating extending two beam spaces beyond the openings, and tapered from thence towards the stringer plate for a distance not less than the breadth of the plating required to be fitted; the thickness of this plating to be the same as given in Table S. 5. for steel decks.

8. Where large openings are adjacent to each other, the intervening space between the openings to be plated over.

9. **Steam Trawlers.**—In all steam trawlers the deck beams should be wholly plated over in way of the engine and boiler casings, and the casings should extend down to the under side of the deck beams and be connected to the deck plating with angle bars and to the half beam ends with angle lugs. If the casings be not extended down to the under side of the beams, they should be attached to the deck plating with angles $4\frac{1}{2} \times 4\frac{1}{2} \times 0{\cdot}36$, having two rows of reeled rivets in each flange. In order to ensure that the scantlings and construction of these casings are satisfactory in every case, detailed plans of the same should be submitted for approval.

## HATCHWAYS.

**Section 32.—7. Connections of Coamings to Deck.**—The coaming plates are to be connected to the deck plating or tie plates with angles of not less thickness than the side coaming plates, welded at the corners of the hatchways. Where a wood deck is fitted, the vertical flanges of the angle bars connecting the side and end coamings to the deck are to extend half an inch above the deck.

8. **Material.**—All hatchway coamings on weather decks and the companions at the fore-end of steamers to be of steel or iron.

9. **Half Beams.**—Where half beams are fitted to alternate frames, they are to be connected to the coaming plates with double angles, and half beams fitted to every frame may be connected to coaming plates,

with single angles of not less thickness than the side coamings. There are to be three rivets in each flange of the angles connecting coamings to the half beams where the depth of the half beam is 7½ inches to 9½ inches, and four rivets where the depth is 10 inches to 12 inches.

**Section 30 (Bunker Hatches, etc.).**—Coal-bunker pipes, where practicable, are to be formed so as to be at least 12 inches above the upper deck, fitted with lids having studs to fit in openings made in the pipes, for their security; the pipes to be so formed that tarpaulin may be securely lashed over them. When there are coal-bunker hatches in the weather deck, they must be properly framed with coaming plates of suitable height, having solid hatches secured by an iron bar or other approved fastening.

**Section 39 (Pumps, etc.).**—1. In steam vessels the pumping arrangements according to the division of holds, &c., to be as follows:—

2. **Holds with double bottoms.**—In the double bottom of each compartment of the hold, and of engine and boiler space, a steam-pump suction is to be fitted at the middle line, and one on each side to clear the tanks of water when the vessel has a heavy list. Where there is considerable rise of floor towards the ends of the vessels, the middle line suction only will be required. A steam-pump suction and a hand-pump are also to be fitted to each bilge in each hold where there is no well. When there is a well, one or three steam-pump suctions are to be fitted in the same, according as there is considerable or little rise of floor, and hand-pumps fitted at the bilges.

3. **Holds without double bottoms.**—Where there is considerable rise of floor, one steam-pump suction and one hand-pump are to be fitted in each hold. Where there is little rise of floor, two or three steam-pump suctions and at least one hand-pump are to be fitted to each hold.

4. **Engine and Boiler space.**—Where a double bottom extends the whole length of engine and boiler space, two steam-pump suctions are to be fitted to the bilge on each side. Where there is a well, one steam-pump suction should be fitted in each bilge and one in the well. Where there is no double bottom in the machinery space, centre and wing steam-pump suctions should be fitted. The rose box of the bilge injection is to be fitted where easily accessible, and is to be used for bilge water only. The main and donkey pumps to draw from all compartments, and the donkey to have also a separate bilge suction in the engine-room.

5. **Fore and After Peaks.**—If the peaks are fitted as water ballast tanks, a separate steam-pump suction is to be led to each. If not used for water ballast, an efficient pump is to be fitted to the fore peak.

6. **Tunnel.**—The tunnel well is to be filled with a steam-pump suction.

7. **All Hand-pumps** to be capable of being worked from the upper or main decks or above the deep load water-line. The pump chambers are not to be more than 24 feet above the suction rose.

**The sizes of the hand-pumps** are to be not less than given in following Table:—

| Tonnage under upper deck. | Hand-pumps in holds. | |
|---|---|---|
| | Diameter of barrel. | Diameter of tail pipes. |
| | Inches. | Inches. |
| In vessels not exceeding 500 tons, . . . | 4 | 2 |
| Above 500 tons but not exceeding 1000 tons, . | 4½ | 2¼ |
| Above 1000 tons but not exceeding 2000 tons, | 5 | 2½ |
| Above 2000 tons, . . . . . . . | 5½ | 2¾ |

In lieu of hand-pumps in each compartment, an approved **fly-wheel pump** may be fitted if it is connected to the steam-pump bilge suction pipes of these compartments.

The hand-pumps may be dispensed with in vessels which have two independent boiler rooms, or a donkey boiler above the bulkhead deck, and steam-pumps (workable from either source of steam) in two separate compartments, connected to the suctions.

8. **No sluice valve or cock** is to be fitted to the collision bulkhead.

9. **No sluice valves or cocks** are to be fitted to the engine-room or other watertight bulkheads unless they are arranged so as to be at all times accessible.

10. **When sluice valves** are fitted, they are to be so arranged as to be controlled above the load water-line, and the rods are to be boxed in to prevent injury.

11. **Sounding pipes** are to be fitted on each side of holds and ballast tanks, and a doubling plate is to be fitted under each.

12. **Air pipes** are to be fitted to each ballast tank as required.

13. All **cocks** and **valves** in connection with bilge and ballast suction pipes are to be fitted in places where they are at all times accessible.

14. All **bilge suction pipes** are to be fitted with strum boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should not be less than double that of the cross section of the suction pipe.

15. The **filling pipes for deep tanks** which can be used for either cargo or ballast must be controlled by valves placed in an accessible position, and so arranged that when the tank is being used for cargo it will be impossible to fill it with water. This result is to be obtained by taking out a short bend or wedge piece and fitting blank flanges in its place, or in some other way to be submitted to and approved by the Committee.

16. The **pipes** for bilge or ballast suctions are to be fitted with flanged joints in convenient lengths, so that they may be easily disconnected for clearing. In the case of cast-iron suction pipes, which

are not also used as tank filling pipes, or which cannot be subjected to sea pressure, spigot and faucet joints made with india-rubber rings fitted over the spigots might be adopted, except in the case of bilge suction pipes passing through ballast tanks, which should be fitted with flanged joints.

17. The **suction pipes** to fore and after peaks, and to the tunnel well, should not be less than 2¼ inches inside diameter, except in vessels not exceeding 500 tons under deck, in which case they may be made 2 inches.

18. The **bilge injection** should not be less than two-thirds of the diameter of the sea inlet to the circulating pump.

19. The **inside diameter** of other **bilge suction pipes** should not be less than given in the following Table :—

### Table CXXII.—Sizes of Bilge Suction Pipes by Lloyd's Rules.

| Tonnage under upper deck. | Engine-room centre suction, separate donkey suction, and hold centre suctions. | Wing suctions in holds where no centre suctions are fitted, and wing suctions in engine-room. | Wing suctions in holds where centre suctions are also fitted. |
|---|---|---|---|
| | Inches. | Inches. | Inches. |
| In vessels not exceeding 500 tons, . . . | 2 | 2 | 2 |
| Above 500 tons but not exceeding 1000 tons, . | 2¼ | 2 | 2 |
| Above 1000 tons but not exceeding 1500 tons, . | 2½ | 2¼ | 2 |
| Above 1500 tons but not exceeding 2000 tons, . | 3 | 2¾ | 2¼ |
| Above 2000 tons but not exceeding 3000 tons, . | 3½ | 3 | 2½ |
| Above 3000 tons, . . | 3½ | 3½ | 2¾ |

In cases where more than one suction to any one compartment are connected to the pumps by a single pipe, this pipe should be not less than the size required for the centre suction.

## LLOYD'S SURVEYS OF MACHINERY.

**Lloyd's ordinary surveys of new machinery** are thus described in the Rules :—

In steam vessels, the machinery and boilers are to be inspected throughout construction, the boilers tested by hydraulic pressure, and the machinery tested under steam by the Society's Engineer-Surveyors, who will furnish a report to the Committee describing them in the manner shown in Form No. 4. If found satisfactory, the Committee will therefore grant a certificate, and insert in the Register Book the notification "LMC." in red (*i.e.* "Lloyd's Machinery Certificate"), indicating that the machinery and boilers are certified to be in good order and safe working condition.

## Lloyd's Periodical Surveys (Section 19).

1. The machinery and boilers of all steam ships and the donkey boilers of sailing vessels are to be surveyed annually if practicable; and in addition are to be submitted to a Special Survey upon the occasions of the vessels undergoing the Special periodical Surveys Nos. 1, 2, and 3, prescribed in the Rules, unless the machinery and boilers have been specially surveyed within a period of twelve months.

2. At these Special Surveys, and on other occasions if deemed necessary by the Surveyors, the propeller, stern-bush, and the sea connections and their fastenings are to be examined while the vessel is in dry dock.

3. The stern shaft is to be examined annually, and drawn and examined at intervals of not more than two years.*

4. The cylinders, pistons, slide valves, crank and tunnel shafts, and pumps are to be examined, and if necessary the condenser is to be examined and tested.

5. The arrangements of cocks, pipes, bilge suctions, roses, &c., are to be examined.

6. The boilers and superheaters are to be examined internally and externally, and if deemed necessary by the Surveyors, both boilers and superheaters are to be drilled or tested by hydraulic pressure; the safe working pressure is to be determined by their actual condition.

7. The safety-valves are to be examined and set to the safe working pressure.

8. If satisfactory, these Surveys will be recorded in the Register Book thus:—"LMC7,08" *in red*, or "B&MS7,08" *in red.*

9. "LMC" (Lloyd's Machinery Certificate) denotes that the machinery and boilers are fitted in accordance with the Rules; and when followed by a date, indicates that they were found at that time to be in good condition. MS, with a date, denotes that the engines at that time were found upon inspection to be in good condition. BS, with a date, denotes that the boilers were found upon inspection at the time to be in good condition.

10. "B&MS" (Boilers and Machinery Surveyed), with a date, denotes that the boilers and machinery, though not fitted strictly in accordance with the rules, were found upon inspection at that time to be in good condition.

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* On the application of owners, the Committee will be prepared to give consideration to the circumstances of any special case.

11. In the event of either the machinery or boilers appearing to be impaired to such an extent as to render it desirable that either or both be specially surveyed within the periods prescribed above, a certificate for either machinery or boilers for a limited period will be granted according to the nature of the case.

### Boilers.

12. The boilers of all steam ships and the donkey boilers of sailing vessels are to be specially surveyed when six years old, and subsequently they are to be specially surveyed annually.

13. At these surveys the boilers and superheaters are to be examined internally and externally, and if deemed necessary by the Surveyors, both boilers and superheaters are to be drilled or tested by hydraulic pressure; the safe working pressure is to be determined by their actual condition.

14. The safety-valves are to be examined and set to the safe working pressure.

15. If satisfactory, these surveys will be recorded in the Register Book thus:—"BS7,08" *in red* in the case of steam vessels, and "DBS7,08" *in red* in the case of sailing vessels.

16. "BS" (Boiler Surveyed) or "DBS" (Donkey Boiler Surveyed), with a date, denotes that the boilers were found upon inspection at that time to be in good condition.

17. In the event of the boilers appearing to be impaired to such an extent as to render it desirable that they be specially surveyed within the periods prescribed above, a certificate for a limited period will be granted according to the nature of the case.

**For Lloyd's requirements as to steel castings,** see under Cast Steel in section on "Materials."

## LLOYD'S RULES RELATING TO SPARE GEAR.

**Lloyd's requirements as to spare gear** are as follows:—

The articles of spare gear mentioned in the following list will be required to be carried in all steam vessels classed in the Society's Register Book, viz.:—

2 connecting-rod, or piston-rod, top end, bolts and nuts.
2 ,, bottom end, bolts and nuts.
2 main bearing bolts.
1 set of coupling bolts.
1 set of feed and bilge pump valves.
1 set of piston springs (where common springs are used).
A quantity of assorted bolts and nuts.
Iron of various sizes.

In addition to the foregoing, the following articles are recommended to be carried with a view to expedite repairs and lessen delay in distant ports, viz. :—

Crankshaft.
Propeller shaft.
Propeller, or a full set of blades.
Stern-bush, or lignum-vitæ lining for bush.
1 pair connecting-rod brasses.
1 pair crosshead brasses.
1 set of check valves.
6 cylinder cover bolts.
6 junk-ring bolts.
4 valve-chest cover bolts.
1 set of link brasses.
1 eccentric strap complete.
Air-pump rod.
Circulating-pump rod.
H.P. valve spindle.
L.P. ,,
2 dozen boiler tubes.
3 ,, condenser tubes.
1 cylinder escape valve and spring.
1 set of safety-valve springs.

## MISCELLANEOUS BOARD OF TRADE RULES.

The following Board of Trade Rules concern the engineer, but cannot properly be placed under any of the preceding section headings:—

79. It is advisable that the donkey engine for pumping water through the condenser of the distilling apparatus be so fitted that it can also be made available in case of emergency for extinguishing fire in any part of the ship; a leather hose, with suitable bends and conductors, should be supplied for this purpose.

80. Each compartment of vessel should have an efficient hand-pump of sufficient size, workable from upper deck. When a double bottom is fitted extending the full length of a compartment and thereby preventing the deck-pump suction being placed near the shell plating at middle line of ship, a separate hand-pump or equivalent must be fitted to draw from waterway at each side of compartment. This regulation need not, however, be enforced in the case of fore and after peaks, chain lockers, or other small compartments near ends of vessel.

Suctions of hand-pumps should, whenever practicable, be placed at after end of compartments in which they are fitted.

When the required deck pumps are not situated on, or attached to, the upper deck, they should be of the closed-top type, and have their discharge pipes carried through the upper deck or through the ship's side, well above the deep-load line. If there are any cocks or valves in connection with the pumps which are not automatic in their action, suitable and efficient appliances should be provided, and kept permanently in position, whereby they may be manipulated from the upper deck.

The above rules as to the fitting of deck pumps with closed tops should be adhered to in new vessels, except in cases where the deck on which the pumps are situated has scuppers for conveying water that may collect upon it to the outside of the vessel and is at such a height above the deep-load line as to preclude the probability of its being submerged in the event of the vessel getting holed. A rose or perforated box of sufficient size should be fitted to each suction pipe, and means should be provided for clearing it and the end of the pipe.

Deck pumps should be provided with suitable handles, and those of the smaller size should have handles long enough for at least two men to work them.

The Surveyor should be careful to see that the deck pumps in large vessels are not placed so high above the bottom of the compartments in which they are situated as to render them inefficient, and if he should at any time have doubts as to their efficiency he should test them.

It is very desirable that the steam winches (if any) be so fitted and arranged as to be available for working the hand deck-pumps.

81. Rotary or fly-wheel pumps may be substituted for ordinary hand-pumps, provided that two such pumps are fitted, and that either is capable of pumping from any hold or machinery compartment in the vessel. They should be worked from a position well above the deep-load line, preferably from the upper deck; and unless they are placed in different watertight compartments, the pump buckets and valves should be capable of being removed from the position of working, so that they can be cleaned and repaired, if necessary, in the event of the compartment in which they are situated being flooded whilst the vessel is fully laden.

If the rotary hand-pumps are connected to the main bilge system, the various bilge suction valves or cocks should be provided with rods carried to a height well above the deep-load line, preferably to the upper deck. Index plates should be fitted at the upper ends to show what each valve or cock is for and how it is operated.

Hand-pumps may be dispensed with, if desired, in vessels fitted with two or more separate watertight boiler-rooms, and a separate watertight engine-room, provided that steam-pumps are fitted in at least two of the compartments named which are not adjacent, and are capable of pumping from any of the bilge suctions in the vessel, and provided also that the steam supply to the pumps is available from the boilers in either or any of the boiler compartments. The arrangement should be such that, in the event of a compartment being flooded, the bilge suction pipes to that compartment can be shut off, and that at least one of the bilge pumps may be available for pumping from any of the other hold or machinery compartments in the vessel.

82. Sounding pipes should be fitted from the upper deck for ascertaining the depth of water in each compartment.

The sounding pipes for the ballast tanks under the engine and boiler-room and under the tunnel floor may, however, be short, provided that they are fitted with screwed caps at the upper ends, or with cocks having the handles secured to the plugs.

83. Suction pipes connected with pumps worked by the main and donkey engines should be carried through the bulkheads into all the

compartments fore and aft of the engine-room, except the compartment in front of the collision bulkhead, so that each compartment can be pumped out separately by the engines. When a double bottom is fitted extending the full length of a compartment, with waterways on each side, a bilge suction should be fitted to each waterway. This requirement need not, however, be enforced in small compartments near the ends of the vessel. The pipes should be well secured where they pass through the bulkheads. In all new steamships, the cocks or valves which are fitted for the purpose of shutting off or controlling the flow of water through these pipes should, unless situated in the compartment occupied by the pump to which the pipes are connected, be provided with means by which they may be manipulated from a height well above the deep-load line, preferably the upper deck.

The arrangement of bilge suction pipes should be such that water cannot pass through them from one compartment to another. Valves in bilge distribution boxes should be of the non-return type, and if cocks are used instead of valves, suitable non-return valves should be fitted in the pipes to prevent water passing from one compartment to another in the event of the cocks being left open.

The free end of each hold suction pipe should be fitted with a suitable rose box or strum, and the tunnel-well suction pipe should be similarly fitted, unless it is provided with a mud-box of the form required for the pipes in connection with the engine-room and stoke-hold bilges.

**Each main and donkey-engine suction pipe** in connection with the engine-room and stoke-hold bilges should be provided with an efficient mud-box, or other similar appliance, placed above the platform, or in any other position in which it will be accessible at all times. The pipe leading from each mud-box to the bilge should, when practicable, be straight, and the cover of the mud-box should be secured in such a manner as to permit of it being expeditiously opened or closed.

**The connection from the donkey engine** to the bilge main should be by means of a switch cock or non-return valve.

**An efficient bilge injection** should be fitted to each main circulating pump.

**A spare tiller** (which has been properly fitted to the rudder-head), relieving tackle, &c., should, in all foreign-going and home-trade steamships, be kept near the after steering gear ready for immediate service. In large steamships where the use of hand gear is impracticable and such gear is not provided, the spare tiller should be attached to the rudder-head ready for immediate use, unless the working tiller is of special design and strength, in which case a spare tiller may not be required, but full particulars should be submitted for the Board's consideration. The steering gear, including chains, should be thoroughly overhauled at every survey, and taken to pieces and thoroughly examined at least once a year. The chains and blocks that are liable to interfere with or endanger the passengers or crew should be guarded by portable, but properly secured, guards.

With the view of relieving, as far as practicable, the rudders of vessels from severe and sudden shocks, springs have in some cases been fitted to the quadrant, or to the rods or chains at each side of the vessel, and the Board think that such fittings, or other efficient means, should be adopted, more particularly in the case of new vessels.

The Surveyors should note that the steam and exhaust pipes of steering engines in all new passenger steamships should be at least of the same internal diameter respectively as the steam and exhaust connections on the cylinders. The arrangement should be such that water will not readily lodge either in the cylinders or in the steam and exhaust pipes. Right-angled bends in the pipes should be avoided as much as possible, and the pipes should be used exclusively for the steering engines. When this is not the case, full particulars and sketches should be submitted to the Board for consideration.

Attention is also directed to a description of steam-steering gear in which a part of the shaft, by which the helmsman actuates the controlling valve, passes through another shaft that is liable to be thrown out of line by the reaction of the spur gearing, and, consequently, is liable to jam the inside shaft to such an extent as to deprive the helmsman of the control of the steering gear. All steam steering gears should be carefully examined, and if any be found constructed in the manner described above, their use should be discouraged, and in any case they should not be approved, unless they have been tested from midship to hard over in both directions, and found satisfactory when the vessel is running at full speed.

It is very desirable that the man at the helm should be so placed that he has a clear lookout ahead, more especially in steamers that frequent crowded harbours or rivers; and, in case of steam or electric launches, the Surveyor must assure himself that view cannot be obstructed by passengers.

77. In passing a helm indicator, the Surveyor should ascertain by actual trial that whenever the pointer moves to the word "port" on the dial or plate, wherever that word may be, it shows that the helm is ported, and whenever the pointer points to the word "starboard," wherever that word may be, it means that the helm is starboarded.

In the foregoing directions it is assumed that "port helm" means that the helm is so moved as to turn the ship's head to the right, and "starboard helm" means that the helm is so moved as to turn the ship's head to the left.

78. Passenger steamers going to sea should be provided with a hose adapted for the purpose of extinguishing fire in any part of the ship, and capable of being connected with the engines of the ship, or with the donkey engine, if it can be worked from the main boiler. The Surveyor must take care that it answers the required purpose.

The fire hose should be connected and stretched, to judge of its length, and thoroughly examined at every survey, and at least once a year (and at any other time that the Surveyor thinks it necessary) it should be tested with the conductor in its place by pumping water through it by the main or donkey engines at full speed. A proper conductor and metal bend or goose neck form part of its equipment, and should be provided. Generally, leather hoses are the most durable, and should be supplied when a declaration for 12 months is required.

191. **Cast-iron stand-pipes, cocks, &c.,** intended for passage through them of hot brine, should not be passed. Surveyors should also discourage the use of cast iron for chocks and saddles for boilers. Particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

200. In the case of steamers performing ocean voyages, and coming in for survey, no question as to gear need be raised if the following spare gear and stores are supplied, or their equivalent, particulars of which should be submitted to the Board for consideration. The heavier portions of this gear should have been fitted and tried in their places, and should be kept on board where access can at all times be had to them:—

1 pair of connecting-rod brasses.
1 air-pump bucket and rod, with guide.
1 circulating-pump bucket and rod.
1 air-pump head valve, seat, and guard.
1 set of india-rubber valves for air-pumps or ⅓ set of metallic ones.
1 circulating pump head valve, seat, and guard.
1 set of india-rubber valves for circulating pumps, or ⅓ set of metallic ones.
2 main bearing bolts and nuts.
2 connecting-rod bolts and nuts.
2 piston-rod bolts and nuts.
8 screw shaft coupling bolts and nuts.
1 set of piston springs suitable for the pistons.
1 set metal feed-pump valves and seats.
3 sets if of india-rubber, or 1 set if of metal, of bilge-pump valves and seats.
1 spring (at least) for each size of escape valve.
1 hydrometer.
Boiler tubes, 3 for each boiler.
100 iron assorted bolts, nuts, and washers, screwed, but need not be turned.
12 brass bolts and nuts, assorted, turned, and fitted.
50 iron ,, ,, ,,
50 condenser tubes.
100 sets of packing for condenser tube ends, or an equivalent.
At least one spare spring of each size for escape valves.
1 set of water-gauge glasses.
$\frac{1}{10}$th of the total number of fire-bars necessary.
3 plates of iron, assorted.
6 bars of iron, assorted.
1 complete set of stocks, dies, and taps, suitable for the engines.
100 iron assorted bolts, nuts, and washers, screwed, but not necessarily turned.

12 brass assorted bolts, nuts, turned and fitted.
50 iron ,, ,, ,,
1 smith's anvil.
1 fitter's vice.
Ratchet braces, and suitable drills.
1 copper or metal hammer.
Suitable blocks and tackling for lifting weights.
1 dozen files, assorted, and handles for the same.
1 set of drifts or expanders for boiler tubes.
1 safety-valve spring (if so fitted) for every four valves if of the same size; but more than 6 spare springs of the same size need not be provided, whatever be the number of the valves; if there are not four valves, then at least one spare spring must be carried.
1 screw jack.
And a set of engineer's tools suitable for the service, including hammers and chisels for vice and forge; solder and soldering iron; sheets of tin and copper; spelter; muriatic acid, or other equivalent, &c.

147. The following list of tools and materials should be provided for distilling apparatus:—

1 set of stoking tools.
1 scaling tool.
1 spanner for boiler doors.
1 set of fire-bars, suitable for boiler.
1 14-inch flat bastard file.
1 14-inch half-round file.
1 10-inch round file.
3 file handles.
2 hand cold chisels.
1 chipping hammer.
1 pair of efficient gas tongs.
1 soldering iron.
10 lbs. of solder.
2 lbs. of rosin.
6 gauge glasses.
24 india-rubber gauge-glass washers.
30 bolts and nuts, assorted.
1 slide rod for donkey pump.
5 lbs. spun yarn.
10 lbs. cotton waste.
1 deal box, with lock, complete.
2 gallons machinery oil.
Animal charcoal sufficient to charge the filter at least twice.
1 can for machinery oil.
1 oil feeder.
1 small bench vice.
1 ratchet brace.
4 drills, assorted.
1 set dies and taps suitable for the bolts.
2 glass salinometers.
1 hydrometer and pot.
1 shifting spanner.
1 lamp for engineer.
And other articles that the particular distiller and boiler supplied may, in the Surveyor's judgment, require.

197. The boiler for supplying steam to the distillers of an emigrant ship should be at least equal in strength to the boilers of passenger steamships, and should have fittings in accordance with these regulations. The steam for working the apparatus is not to be taken from the main boilers. No exhaust steam may be permitted to go into the

condenser, if appliances for the introduction of lubricants are fitted to the steam pipes or steam cylinder of the pumping engine. The boiler of the apparatus must not be filled or fed with water from the surface condensers of the main engines, and must not be fitted with cocks, &c., for the introduction of tallow or oil. The presence of zinc in such boilers is also objectionable, and the Surveyors should refuse to pass boilers so fitted for use on board vessels clearing as emigrant ships.

No distilling apparatus should be passed unless fitted with a filter of suitable size charged with animal charcoal.

When the water is pumped into the condenser, the latter should be fitted with an efficient escape-valve which cannot readily be tampered with ; and, if the condensing portions of the apparatus or the cooler and filter are unfit to bear the pressure on the boiler, an efficient safety-valve which cannot readily be overloaded, should be fitted between the steam pipe and the apparatus.

The Surveyor should satisfy himself as to the capability of the man who is to have charge of the apparatus.

As the Surveyor will be held wholly responsible for the efficiency of the distillers, it rests with him to decide whether or not the apparatus should, in the case of emigrant vessels furnished with passenger certificates, be taken to pieces for examination prior to every voyage, but the distilling apparatus of such vessels must be taken to pieces for examination at least once every twelve months, or more often if the Surveyor thinks it necessary. The tubes or coils should be tested to at least twice the load on the safety-valve on the apparatus, or, in cases where no safety-valve is fitted, to twice the highest working pressure of the boiler from which the apparatus can be worked, and the machinery and boiler should be thoroughly examined. The charcoal should be removed from the filter, cleansed, or renewed, at least once every six months.

After the distilling apparatus is put together again, it should be tested as to the quantity and quality of the water made, and this should also be done before the commencement of every voyage. The water should be cool, pure, and fit to drink immediately it is drawn off from the filter.

The presence of zinc in such boilers is also objectionable, and Surveyors should refuse to pass boilers so fitted for use on board vessels sailing under the Merchant Shipping Act of 1894, Part III.

When the water is pumped into the condenser there should be an efficient escape-valve on the condenser, which cannot be readily tampered with, and if the condensing portion of the apparatus or the cooler and filter be unfit to bear the pressure on the boiler, an efficient safety-valve that cannot be readily overloaded should be fitted between the steam pipe and the apparatus.

146. It is advisable that the donkey engine for pumping water through the condenser be so fitted that it can be made available in case of emergency for extinguishing fire in any part of the ship ; a leather hose, with suitable bends and conductors, should be applied for this purpose.

## CHAINS AND ROPES

The following Tables give the Admiralty requirements as to chains:—

### Table CXXIII. Admiralty Tests, &c., of Stud-link Chain Cable.

| Dia. of cable in inches. | Breaking strength in tons. | Proof load in tons. | Weight of 100 fathoms in cwts. | Dia. of cable in inches. | Breaking strength in tons. | Proof load in tons. | Weight of 100 fathoms in cwts. |
|---|---|---|---|---|---|---|---|
| 7/16 | 4·90 | 3·5 | 9·25 | 1½ | 56·70 | 40·5 | 108·0 |
| ½ | 6·30 | 4·5 | 12·0 | 1⅝ | 66·50 | 47·5 | 126·75 |
| 9/16 | 7·70 | 5·5 | 15·25 | 1¾ | 77·17 | 55·125 | 147·0 |
| ⅝ | 9·80 | 7·0 | 18·75 | 1⅞ | 88·55 | 63·25 | 168·75 |
| 11/16 | 11·90 | 8·5 | 22·75 | 2 | 100·80 | 72·0 | 192·0 |
| ¾ | 14·17 | 10·125 | 27·0 | 2⅛ | 113·75 | 81·25 | 216·75 |
| ⅞ | 19·25 | 13·75 | 36·75 | 2¼ | 127·57 | 91·125 | 243·0 |
| 1 | 25·20 | 18·0 | 48·0 | 2⅜ | 142·10 | 101·5 | 270·75 |
| 1⅛ | 31·85 | 22·75 | 60·75 | 2½ | 157·50 | 112·5 | 300·0 |
| 1¼ | 39·37 | 28·125 | 75·0 | 2¾ | 181·02 | 129·3 | 363·0 |
| 1⅜ | 47·60 | 34·0 | 90·75 | 3 | 204·12 | 145·8 | 432·0 |

Up to and including 2½ inches diameter, the above proof loads are equal to 630 lbs. per circular ⅛ inch of section of one side of link, and are the same as those required by the Chain Cables and Anchors Acts; but for 2¾ inches diameter the load is reduced to 598·5 lbs. per circular ⅛ inch, and for 3 inches diameter to 576 lbs. The formula for proof loads of chains up to 2½ inches diameter may also be written,—

**Proof load in tons = 18 × (diameter in inches)$^2$.**

The breaking strengths are placed at 40 per cent. above the proof loads.

### Table CXXIIIa.—Admiralty Tests, &c., of Short-link Chain.

| Dia. of chain in inches. | Breaking strength in tons. | Proof load in tons. | Weight per fathom in pounds. | Dia. of chain in inches. | Breaking strength in tons. | Proof load in tons. | Weight per fathom in pounds. |
|---|---|---|---|---|---|---|---|
| ¼ | 1·87 | ·75 | 3·0 | 15/16 | 26·37 | 10·55 | 49·0 |
| 5/16 | 2·93 | 1·17 | 5·5 | 1 | 30·00 | 12·00 | 56·0 |
| ⅜ | 4·22 | 1·69 | 8·0 | 1 1/16 | 33·87 | 13·54 | 63·0 |
| 7/16 | 5·74 | 2·30 | 10·5 | 1⅛ | 37·97 | 15·18 | 71·0 |
| ½ | 7·50 | 3·00 | 14·0 | 1 3/16 | 42·30 | 16·92 | 79·0 |
| 9/16 | 9·49 | 3·80 | 18·0 | 1¼ | 46·87 | 18·75 | 87·0 |
| ⅝ | 11·72 | 4·69 | 22·0 | 1 5/16 | 51·68 | 20·67 | 96·0 |
| 11/16 | 14·18 | 5·67 | 27·0 | 1⅜ | 56·72 | 22·68 | 106·0 |
| ¾ | 16·87 | 6·75 | 32·0 | 1 7/16 | 62·00 | 24·80 | 116·0 |
| 13/16 | 19·80 | 7·92 | 37·0 | 1½ | 67·50 | 27·00 | 127·0 |
| ⅞ | 22·97 | 9·19 | 43·0 | ... | ... | ... | ... |

The proof loads for short-link chains are ⅔rds those for stud-link chains.

The rule may be written,—**Proof load in tons = 12 × (diameter in inches)$^2$.** The breaking strengths of short-link chains are placed at 2½ times the proof loads.

Lloyd's requirements as to chain cables, &c., are given in Table CXXIV.; the proof loads and breaking strengths are those required by the Act of Parliament; the breaking strengths for chains above 1½ inches diameter are the same as required by the Admiralty, but for chains of 1½ inches diameter and under, Lloyd's require a slightly higher breaking strength.

### Table CXXIV.—Lloyd's Tests of Stud-link Chain Cables.

| Diameter of chain in inches. | Proof load (Statutory) in tons. | Breaking strength (Statutory) in tons. | Diameter of chain in inches. | Proof load (Statutory) in tons. | Breaking strength (Statutory) in tons. |
|---|---|---|---|---|---|
| $\frac{11}{16}$ | 8·5 | 12·75 | 1⅝ | 47·5 | 66·5 |
| ¾ | 10·125 | 15·125 | $1\frac{11}{16}$ | 51·25 | 71·75 |
| $\frac{13}{16}$ | 11·875 | 17·8 | 1¾ | 55·125 | 77·125 |
| ⅞ | 13·75 | 20·625 | $1\frac{13}{16}$ | 59·125 | 82·75 |
| $\frac{15}{16}$ | 15·8 | 23·7 | 1⅞ | 63·25 | 88·5 |
| 1 | 18·0 | 27·0 | $1\frac{15}{16}$ | 67·5 | 94·5 |
| $1\frac{1}{16}$ | 20·3 | 30·4 | 2 | 72·0 | 100·8 |
| 1⅛ | 22·75 | 34·125 | $2\frac{1}{16}$ | 76·5 | 107·1 |
| $1\frac{3}{16}$ | 25·375 | 38·0 | 2⅛ | 81·25 | 113·75 |
| 1¼ | 28·125 | 42·125 | $2\frac{3}{16}$ | 86·125 | 120·5 |
| $1\frac{5}{16}$ | 31·0 | 46·5 | 2¼ | 91·125 | 127·5 |
| 1⅜ | 34·0 | 51·0 | $2\frac{5}{16}$ | 96·25 | 134·75 |
| $1\frac{7}{16}$ | 37·125 | 55·625 | 2⅜ | 101·5 | 142·1 |
| 1½ | 40·5 | 58·7 | $2\frac{7}{16}$ | 106·9 | 149·625 |
| $1\frac{9}{16}$ | 43·9 | 61·4 | 2½ | 112·5 | 157·5 |

*Note.*—Unstudded close-link chains will be admitted as cables if proved to two-thirds the load required for stud-link chains, and if the breaking strength is not less than twice such proof load.

In some recent tests of the chain cables of large steamers a 2⅝-inch cable gave an average ultimate strength of 212 tons, and a $2\frac{9}{16}$-inch cable gave 223 tons as the lowest and 229 tons as the highest of seven tests.

**The safe working load on chains** should not be taken higher than half the proof load; if this proportion be adopted the formula becomes:—

$$\textbf{Working load in tons} = \begin{cases} 9 \times (\text{diameter in inches})^2 \text{ for stud-link.} \\ 6 \times (\text{diameter in inches})^2 \text{ for close-link.} \end{cases}$$

For ordinary crane chains and slings $4 \times (\text{diameter in inches})^2$ is high enough.

## Table CXXV.—Admiralty Flexible Steel Wire Ropes.

| Size of rope (circumference) in inches. | Number of wires in each strand. | Weight per fathom in lbs. | Minimum breaking st'gth in tons. | Torsion test; No. of twists each wire must stand. | Ductility tests, &c. |
|---|---|---|---|---|---|
| 8 | 30 | 53 | 148 | 9 | Each rope to consist of 6 strands. Wires to be of best crucible steel galvanized with pure zinc.<br>The rope is to be laid up evenly and uniformly as regards size and angle, and is to contain a proper sized hemp core.<br>A latitude not exceeding 5 per cent. over or under the prescribed weights will be allowed.<br>Elongation will be assumed to commence when one-sixth of the breaking-load has been applied.<br>**Test A** (torsion).—Each wire to stand being twisted through the number of revolutions stated in column 5, in a length of 8 inches.<br>**Test B** (bending).—Each wire to stand coiling around itself eight turns and back again.<br>If, in being tested, one wire in seven fail, but the other six give fair and uniform results, the average being up to the standard, the rope will be considered satisfactory in that respect. |
| 7 | 30 | 41 | 113 | 11 | |
| 6½ | 30 | 35 | 98 | 14 | |
| 6 | 30 | 31 | 84 | 15 | |
| 5½ | 24 | 28 | 71 | 16 | |
| 5 | 24 | 23 | 59 | 17 | |
| 4½ | 12 | 14 | 39 | 15 | |
| 4 | 12 | 12 | 31 | 17 | |
| 3½ | 12 | 9 | 24 | 18 | |
| 3 | 12 | 7 | 17 | 22 | |
| 2¾ | 12 | 5½ | 14½ | 25 | |
| 2½ | 12 | 4½ | 11¾ | 26 | |
| 2¼ | 12 | 3¾ | 9 | 28 | |
| 2 | 12 | 2¾ | 7 | 33 | |
| 1¾ | 12 | 2 | 5½ | 36 | |
| 1½ | 12 | 1¾ | 4 | 41 | |
| 1¼ | 12 | 1¼ | 2⅞ | 47 | |
| 1 | 12 | ¾ | 1¾ | 60 | |

**Proportions of Links of Chains.**—The standard proportions of the links of chains, in terms of the diameter of the bar from which they are made, are as follows:—

| | Overall length. | Overall breadth. |
|---|---|---|
| Stud-link | 6 diameters | 3·6 diameters |
| Close-link | 5 ,, | 3·5 ,, |

The stud has usually a diameter at the centre of ·6 × diameter of chain, and at the ends 1 × diameter of chain.

**Weight of Chains.**—The weight of stud-link chain cables is given very nearly by the rule,—

$$W = 55d^2,$$

where $W$ = weight per fathom in pounds, and $d$ = diameter of bar from which chain is made.

For close-link chain the rule becomes,—

$$W = 58d^2.$$

The above rules give weights about midway between those required by the Admiralty and those required by Lloyd's. The weight is of course much affected by the length of link used.

## Steel Wire Ropes.

The Admiralty requirements with regard to flexible steel wire ropes are given in Table CXXV.

Steel wire ropes for standing rigging are required to be made of fewer wires of larger diameter, are rather heavier, and must be of rather greater ultimate strength.

The following Table shows the breaking strengths that steel wire hawsers, &c., must show in order to be accepted by Lloyd's:—

### Table CXXVI.—Breaking Strengths of Steel Wire Hawsers, &c., by Lloyd's Rules.

| Size (circumference) in inches. | Breaking strength in tons. | Size (circumference) in inches. | Breaking strength in tons. | Size (circumference) in inches. | Breaking strength in tons. |
|---|---|---|---|---|---|
| 2 | 7 | 3¾ | 29 | 5½ | 71 |
| 2¼ | 9½ | 4 | 33 | 5¾ | 78 |
| 2½ | 12¼ | 4¼ | 35 | 6 | 85 |
| 2¾ | 15½ | 4½ | 39 | 6½ | 100 |
| 3 | 18 | 4¾ | 47 | 7 | 113 |
| 3¼ | 22 | 5 | 59 | 7½ | 128 |
| 3½ | 26 | 5¼ | 65 | 8 | 149 |

## Table CXXVIa.—Special Flexible Steel Wire Rope.

| Size (circumference) in inches. | Lloyd's Breaking Test in tons. | Size (circumference) in inches. | Lloyd's Breaking Test in tons. |
|---|---|---|---|
| 1¾ | 8·9 | 3¾ | 41·0 |
| 2 | 11·7 | 4 | 52·5 |
| 2¼ | 14·6 | 4¼ | 59·0 |
| 2½ | 18·2 | 4½ | 65.5 |
| 2¾ | 22·0 | 5 | 73·0 |
| 3 | 26·2 | 5½ | 88·0 |
| 3¼ | 30·7 | 6¼ | 114 |
| 3½ | 35·5 | | |

A short length of each of the wires composing the hawser, &c., will also be required, after being galvanised, to show a tensile strength equivalent to that given in the above table, and the aggregate strength of the wires must not be less than 10 per cent. in excess of that strength.

Each wire must also be capable of being twisted around itself not less than eight times, and of being untwisted and straightened again without breaking.

The strength of steel wire ropes, relatively to their girth, depends not only on the quality of the wire used, but also on the amount of hemp core used; usually there is a central hemp core, and sometimes each strand has also a similar core, but sometimes there are no hemp cores at all. There would be no particular difficulty in obtaining ropes to stand twice the tests given in the above Table, but such ropes would probably be wanting in flexibility, and would require drums of very large diameter if they were to work satisfactorily and to last any time.

### Hemp Ropes.

Hemp is laid up right-handed in yarns; and yarns are laid up left-handed into strands.

A hawser is composed of three strands laid up right-handed.

A cable is composed of three hawsers laid up left-handed.

Shroud-laid rope has a core surrounded by four strands.

The strength of hemp ropes depends on the quality of the hemp used, on the type or make of the rope, and on its condition (*i.e.* wet or dry, tarred or untarred).

The twist diminishes the strength, but increases the solidity and durability, and the strength therefore depends to some extent on the twist.

When a rope is wet or tarred its strength is reduced by about one-fourth.

The working strength is commonly taken as ⅛th of the breaking strength.

## Table CXXVII.—Bullivant's Steel Wire Ropes (Galvanised)

For Ships' Hawsers, Running Rigging, Cargo Falls, etc.

TABLE SHOWING WEIGHTS AND BREAKING STRESSES.

| Flexible Steel Wire Rope, 6 Strands, each 12 Wires. | | | | | Extra Flexible Steel Wire Rope, 6 Strands, each 24 Wires. | | Special Extra Flexible Steel Wire Rope. | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | 6 Strands, each 37 Wires. | | Special Make. |
| Size Circumference. | Size Diameter. | Approx. Weight per Fathom | Approx. Breaking Stress (Actual). | Diamr. of Barrel or Sheave round which it may be worked at a slow speed. | Approx. Weight per Fathom | Approx. Breaking Stress (Actual). | Approx. Weight per Fathom. | Approx. Breaking Stress (Actual). | Approx. Breaking Stress (Actual). |
| Ins. | Ins. | Lbs. | Tons. | Ins. | Lbs. | Tons. | Lbs | Tons. | Tons. |
| ¾ | ¼ | ·420 | 1·1 | 4·5 | ·488 | 1·5 | ... | ... | ... |
| 1 | 5/16 | ·720 | 2·0 | 6 | ·960 | 2·8 | 1·08 | 3·0 | ... |
| 1¼ | ... | 1·08 | 3·1 | 7·5 | 1·44 | 4·3 | 1·50 | 4·0 | ... |
| 1½ | ... | 1·50 | 4·4 | 9 | 1·98 | 6·0 | 2·16 | 6·6 | ... |
| 1¾ | 9/16 | 2·10 | 5·9 | 10·5 | 2·82 | 8·5 | 3·00 | 8·5 | ... |
| 2 | ⅝ | 2·70 | 8·0 | 12 | 3·60 | 10·9 | 3·96 | 11·3 | ... |
| 2¼ | ... | 3·60 | 10·5 | 13·5 | 4·68 | 14·2 | 5·04 | 14·5 | ... |
| 2½ | 13/16 | 4·20 | 12·3 | 15 | 5·70 | 17·2 | 6·12 | 18·2 | ... |
| 2¾ | ⅞ | 5·10 | 14·8 | 16·5 | 6·78 | 20·5 | 7·38 | 22·2 | ... |
| 3 | 15/16 | 6·00 | 17·6 | 18 | 8·04 | 25·0 | 8·70 | 26·6 | ... |
| 3¼ | ... | 7·30 | 21·1 | 19·5 | 9·54 | 28·9 | 10·32 | 29·0 | ... |
| 3½ | 1⅛ | 8·40 | 24·4 | 21 | 11·28 | 34·2 | 11·88 | 34 0 | ... |
| 3¾ | .. | 9·60 | 27·8 | 22·5 | 12·78 | 38·8 | 13·80 | 39·5 | ... |
| 4 | ... | 10·98 | 32·3 | 24 | 14·34 | 43·6 | 15·78 | 45·3 | ... |
| 4¼ | ... | 12·00 | 35·4 | 25·5 | 16·50 | 50·1 | 17·70 | 51·6 | ... |
| 4½ | ... | 14·04 | 41·3 | 27 | 18·72 | 57·0 | 19·98 | 58·2 | ... |
| 4¾ | 1½ | ... | ... | ... | 20·64 | 62·8 | 22·14 | 65·3 | ... |
| 5 | ... | ... | ... | ... | 22·68 | 68·9 | 24·54 | 72·7 | ... |
| 5¼ | ... | ... | ... | ... | 24·78 | 75·3 | 27·18 | 80·6 | ... |
| 5½ | 1¾ | ... | ... | ... | 27·00 | 82·0 | 29·58 | 86·7 | ... |
| 5¾ | ... | ... | ... | ... | ... | ... | 32·28 | 93·1 | ... |
| 6 | 1⅞ | ... | ... | ... | ... | ... | 35·40 | 104·3 | ... |
| 6½ | ... | ... | ... | ... | ... | ... | 40·92 | 118·5 | ... |
| 7 | ... | ... | ... | ... | ... | ... | 47·94 | 136·2 | ... |
| 7½ | ... | ... | ... | ... | ... | ... | 55·00 | 178·5 | ... |
| 8 | ... | ... | ... | ... | ... | ... | 63·00 | 198·0 | 202 |
| 9 | ... | ... | ... | ... | ... | ... | 79·00 | 250·0 | 257 |
| 10 | ... | ... | ... | ... | ... | ... | 98·00 | 305·0 | 318 |
| 11 | ... | ... | ... | ... | ... | ... | 120·00 | ... | 381 |
| 12 | ... | ... | ... | ... | ... | ... | 142·00 | ... | 455 |

In these Flexible Rope Tables the wire is calculated as taking a breaking stress of 90 tons to the square inch.

Messrs BULLIVANT & CO., LTD., are prepared to make steel ropes to take a *breaking stress of* 1200 *tons on the single part*, and they will furnish any information connected with larger size cables than those mentioned in the above Table.

## Table CXXVIII.—Bullivant's Mild Plough Steel Wire Crane Ropes (Black).

| Flexible Steel Wire Rope, 6 Strands, each 12 Wires. | | | | Extra Flexible Steel Wire Rope, 6 Strands, each 24 Wires. | | Special Extra Flexible Steel Wire Rope, 6 Strands, each 37 Wires. | |
|---|---|---|---|---|---|---|---|
| Size Circumference. | Approx. Diam. | Approx. Weight per Fathom. | Approx. Breaking Stress. | Approx. Weight per Fathom. | Approx. Breaking Stress. | Approx. Weight per Fathom. | Approx. Breaking Stress. |
| Ins. | Ins. | Lbs. | Tons. | Lbs. | Tons. | Lbs. | Tons. |
| 1 | $\frac{5}{16}$ | ·8 | 1·9 | ·9 | 3·0 | 1·1 | 3·2 |
| $1\frac{1}{4}$ | ... | 1·1 | 2·9 | 1·5 | 5·0 | 1·6 | 4·8 |
| $1\frac{1}{2}$ | ... | 1·6 | 4·1 | 2·0 | 7·0 | 2·4 | 7·1 |
| $1\frac{3}{4}$ | $\frac{9}{16}$ | 2·1 | 5·6 | 2·7 | 9·3 | 3·2 | 9·9 |
| 2 | $\frac{5}{8}$ | 2·9 | 7·6 | 3·7 | 12·7 | 4·3 | 13·2 |
| $2\frac{1}{4}$ | ... | 3·6 | 9·5 | 4·5 | 15·8 | 5·2 | 17·0 |
| $2\frac{1}{2}$ | $\frac{13}{16}$ | 4·4 | 11·7 | 5·5 | 19·2 | 6·4 | 20·1 |
| $2\frac{3}{4}$ | $\frac{7}{8}$ | 5·3 | 14·0 | 6·9 | 23·9 | 7·9 | 24·7 |
| 3 | $\frac{15}{16}$ | 6·2 | 16·6 | 8·0 | 28·1 | 9·2 | 29·8 |
| $3\frac{1}{4}$ | ... | 7·5 | 20·0 | 9·7 | 33·8 | 11·0 | 33·8 |
| $3\frac{1}{2}$ | $1\frac{1}{8}$ | 8·6 | 23·0 | 11·1 | 38·7 | 12·9 | 39·7 |
| $3\frac{3}{4}$ | ... | 9·8 | 26·3 | 12·5 | 43·9 | 14·8 | 46·0 |
| 4 | ... | 11·1 | 29·8 | 14·5 | 50·9 | 16·5 | 52·9 |
| $4\frac{1}{4}$ | ... | 12·5 | 33·6 | 16·2 | 56·9 | 18·7 | 60·1 |
| $4\frac{1}{2}$ | ... | 14·2 | 38·2 | 18·0 | 63·2 | 21·0 | 65·9 |
| $4\frac{3}{4}$ | $1\frac{1}{2}$ | 15·8 | 42·4 | 20·4 | 71·6 | 23·5 | 74·1 |
| 5 | ... | 17·4 | 46·8 | 22·4 | 78·7 | 25·7 | 82·6 |

*Note.*—The diameter of barrel or sheave for flexible ropes should not be less than six times the circumference of the rope, but where extra flexible and special extra flexible ropes are used the diameter may be somewhat reduced.

The Admiralty requirements as to various sizes of hawser-laid cordage in common use are given in Table CXXIX. ; the 11-inch rope is specified to be of tarred Petersburg hemp, but all the smaller sizes are to be of tarred Riga hemp ; all are to be three-strand.

**Table CXXIX.—Admiralty Tarred Hemp Cordage.**

| Size of rope (circumference) in inches. | Size of yarn. | Number of threads in the rope. | Standard breaking strength. | | |
|---|---|---|---|---|---|
| | | | Tons. | cwt. | qrs. |
| ½ | 40 | 6 | 0 | 3 | 0 |
| ¾ | ,, | 12 | 0 | 6 | 0 |
| 1 | ,, | 15 | 0 | 8 | 0 |
| 1½ | ,, | 33 | 0 | 15 | 0 |
| 1¾ | ,, | 42 | 1 | 0 | 0 |
| 2 | ,, | 54 | 1 | 7 | 0 |
| 2½ | ,, | 84 | 2 | 0 | 0 |
| 3 | ,, | 120 | 3 | 0 | 0 |
| 3½ | 30 | 123 | 3 | 18 | 0 |
| 4 | ,, | 159 | 5 | 0 | 0 |
| 4½ | ,, | 201 | 6 | 9 | 0 |
| 5 | ,, | 249 | 7 | 18 | 0 |
| 11 | 25 | 1008 | 36 | 10 | 0 |

All sizes are specified to be formed at an angle of 27°, hardened at 37°, and finished at 42°.

If the smallest size be excluded, the above Table gives breaking strengths which approximate very closely to those given by the formula,—

$$\textbf{Breaking strength in cwts.} = (6{\cdot}2 \times \textbf{girth in inches}^2) + 2.$$

The weight in pounds per fathom of hawser-laid hemp ropes is given approximately by the rule :—

Weight in lbs. per fathom $= {\cdot}17 \times (\text{girth in inches})^2$ for dry ropes.
,, ,, $= {\cdot}21 \times (\text{girth in inches})^2$ for wet or tarred ropes.

## STRENGTH, &c., OF MATERIALS.

Generally, where reference is made in this section to the extension per cent. of a test piece, it is to be understood that the piece is 8 diameters long between the marks, if of circular section, or 12·75 thicknesses long by 2 in width, if of rectangular section.

## Cast Iron.

The strength that an iron casting may be expected to possess depends on the quality of the iron, on the number of times it has been melted, on the design or form of the casting, and on the skill and care exercised in the foundry to ensure soundness and freedom from contraction stresses. The quality of the iron depends on its chemical constitution, and on the method of manufacture. The effects of variations in chemical composition are indicated in the following table :—

### Table CXXX.—Compositions and Qualities of Cast Iron.

| Quality. | Composition. | | |
|---|---|---|---|
| | Combined carbon per cent. | Graphitic carbon per cent. | Silicon per cent. |
| Very soft, . . . | ·15 | 3·1 | 2·5 |
| Very hard, . . . | ... | ... | under ·8 |
| Great general strength, . | ·5 | 2·8 | 1·42 |
| Great tensile strength, . | ... | ... | 1·8 |
| Great crushing strength, | over 1·0 | under 2·6 | about ·8 |

Pig irons are as a rule divided into grades, in number sometimes seven but often only three, each of which is known by a number. No. 1, grey iron, contains the most free or graphitic carbon, and, when broken, exhibits a very coarsely granular fracture with dark grey scales of considerable size; when melted, "it runs very thin," or is of extreme fluidity, and is therefore used mostly for fine ornamental castings, and for mixing with other numbers when increased fluidity is required.

The highest number, whether there be seven or three grades, is a *white iron*, and contains the carbon in a combined state with little or none free. The middle number or numbers have the carbon partly free and partly combined, and called *mottled iron* from the appearance of fracture.

When in seven grades, No. 2 pig is neither so soft nor so fluid as No. 1, but is not sufficiently close grained for general use.

No. 3 pig is that usually employed for marine engine castings ; by adding No. 1 a mixture suitable for smaller complicated castings is obtained, and the addition of No. 4 gives a harder and closer-grained metal.

No. 4 is almost a forge iron, not much used in the foundry except for such mixing purposes ; it still shows a mottled grey fracture, but the grain is finer and more crystalline, and there is an absence of the graphitic scales so marked in Nos. 1, 2, and 3.

Nos. 5 and 6 are not used in the foundry at all, but are made for conversion into wrought iron, &c.

No. 7 pig shows a silvery white and crystalline fracture, contains practically no free carbon, and is extremely hard. Like Nos. 5 and 6, it is a "forge iron," and is sometimes called "white forge," while Nos. 5 and 6 are called "grey forge."

Iron from different districts, and made from different classes of ore, of course varies considerably in composition and quality, but quality is also considerably affected by the method of reduction,—cold-blast iron being usually stronger, tougher, and closer grained than hot-blast, and therefore often used for mixing with other irons where exceptional strength and toughness is required.

**Iron Mixtures.**—As all cast irons are improved by mixing as well as re-melting, no important casting should be made entirely of new pig, and if maximum strength is required, the whole of the material should be re-melted. Chemical analysis should be made and the same composition aimed at by judicious mixing to get uniformly good results.

For cylinders a strong, tough, and close-grained metal is required, which may be obtained by using equal proportions of picked scrap, best Scotch No. 3 pig, and good cold blast. If the cylinder is to have liners and false faces, most of the hardening elements may be reduced.

Cylinder liners and false faces require a good amount of strength and great hardness, and may be made of a similar mixture hardened up with some No. 4 pig.

Since cold-blast irons are scarce, and always dear, hematite with good Scotch or other equally suitable irons may be used for such purposes. In fact, hematite is often used with No. 1 and No. 3 irons of quite ordinary brands for cylinders, &c., of cargo-boat engines when weight is not of prime importance.

Castings of simple form, such as propeller blades and bosses, may be increased in hardness, strength, and closeness of grain by the addition of steel boiler plate scrap to the extent of 10 per cent. or even higher. Hematite is used also for the same purpose.

**The contraction of iron castings** in cooling varies considerably with the form and proportions of the casting, but is, on an average, $^1/_{10}$ to ⅛ inch per foot.

The cooling of large and intricate castings should be as gradual as possible, as the internal stresses are then somewhat relieved, and the risk of cracks diminished. The slower the cooling is the larger will the crystals be, and *vice versa*, but slow cooling becomes a kind of annealing.

**The weight of cast iron** also varies very considerably,—the difference between the heaviest and lightest kinds being nearly 40 lbs. per cubic foot; but a fair average value, and one easily remembered, is about 450 lbs. per cubic foot, or ·26 lbs. per cubic inch. A plate 1 foot square × 1 inch thick will then weigh 37·5 lbs. Ordinary marine castings are probably rather above than below this weight.

**Strength of Cast iron.**—Cast iron suitable for ordinary marine castings should not have a lower ultimate tensile strength than 17,000 lbs. per square inch, and when weight is of importance, and scantlings are cut down, a strength of 20,000 to 22,000 lbs. per square inch should be aimed at, and even 28,000 can be attained with judicious mixing and treatment.

Testing, both chemical and mechanical, should be constantly resorted to if there is to be any check on the foundry, or if any accurate knowledge as to the material being turned out is required.

The ultimate strength of cast iron in compression is about 90,000 lbs. per square inch; in ordinary construction it may carry three times as much as in tension.

**The Admiralty requirements as to cast iron** are as follows:—

Test pieces to be taken from such castings as the inspecting officer may consider necessary. The minimum tensile strength to be 9 tons (20,160 lbs.) per square inch, taken on a length of not less than 2 inches.

The transverse breaking load for a bar 1 inch square, loaded at the middle between supports 1 foot apart, is to be not less than 2000 lbs.

## Wrought Iron.

The quality of wrought iron,—provided it is free from such harmful ingredients as sulphur and phosphorus,—depends largely on the amount of work that has been done on it at the mill,—*i.e.* on the extent to which it has been rolled down and the so-called fibre developed.

The appearance of the fracture depends a good deal on the manner in which it is broken; if good bar iron is nicked at one side, and slowly broken or bent back, it should show a clear white silvery, and almost entirely fibrous, fracture, whereas if nicked on two sides, or broken with a smart severe blow, the fracture will show more white crystalline grain, with little or no fibre.

A specimen from a good forging will show a clear and silvery grain, but larger than that of the bar iron, with about 20 to 30 per cent. of fibrous patches.

An inferior iron usually shows a coarse crystalline structure, or if fibre is present, it is dull and earthy looking.

Coarse crystals, or large shining plates, generally indicate a "cold-short" iron, and "red-shortness" is indicated by an earthy, dull, or dark fracture.

**Merchant bar** is the commonest quality formerly used for ladders, gratings, fire-bars, bearer bars, &c., but steel is now generally cheaper and much better for these purposes.

**Best bar** is the next quality; its tensile strength is about 24 tons (or, say, 54,000 lbs.) per square inch, and it may be used for all ordinary smithing purposes.

**Best best bar** is a higher quality again, and has an ultimate tenacity of 26 to 27 tons (or 58,000 to 60,000 lbs.) per square inch, with an elongation of about 25 per cent. in 8 inches, and a contraction of area of about 50 per cent.; the fibre is uniform and silky in appearance, and the bar may be bent double, cold, without fracture.

It is still largely used for the screwed stays of boilers,* as it stands better than steel the strains coming on them; the Board of Trade permit the use with a working stress of 9000 lbs. per square inch, providing the iron is tested as is steel. Bar steel has largely taken the place of iron, but for smithing purposes or for chain-making where welding is resorted to, best Yorkshire and Staffordshire bars, and bars made from boiler plate scrap, remain still in quite good demand.

**Weight of wrought iron.**—Wrought iron varies slightly in density according to the method of manufacture, the form into which it is put, and the amount of impurity it contains. A fair average value is 485 lbs. per cubic foot, or ·28 lb. per cubic inch; bars and Yorkshire plates give about this figure, but Staffordshire plates are rather lighter (about 480 lbs.), and large forgings lighter again (about 477 lbs.),—owing probably to the presence of cinder in a minutely divided condition. A square foot of plate, one inch thick, usually weighs 40 lbs.

## Cast Steel.

**Steel castings** for the pistons, covers, framing, &c., of marine engines should have an ultimate tensile strength of 29 to 31 tons (65,000 to 69,000 lbs.) per square inch, with an extension of about 12 to 15 per cent in 8 inches. For castings that are at all intricate, or thin in parts, it is necessary to use a steel with more carbon and a strength up to 36 or 38 tons ultimate, as the milder and tougher steels are not sufficiently fluid when melted, but then the extension obtainable will often not exceed 8 per cent. The ultimate tensile in tons per square inch added to extension per cent. should be not less than 45. British Engineering Standard is 50.

**The contraction of cast steel** in cooling is more variable than that of cast iron, but is on an average about $^3/_{16}$ inch per foot, or the same as that of brass. Special care should therefore be taken in designing large pieces to be cast in this metal, and forms that will interfere with the contraction, or cause the casting to "draw" in cooling should be avoided; when possible an open or H section should be preferred to a close or box section for framings, as the former is more likely to give a sound casting,—especially if plenty of small fillets are placed in the angles (say ⅜ inch thick and 6 inches pitch—for large framings—dying away at 3 inches out from the angle in both directions) and a good radius is used.

**Annealing**, for the purpose of relieving internal stresses set up, is absolutely necessary for large or intricate castings, and desirable for all.

Soft steels of this class will contain up to ·3 per cent. of carbon; when the proportion reaches about 1 per cent. it becomes possible to harden or temper the steel.

Good hard cast steel, with an ultimate strength of 50 or 55 tons, will probably contain 1·3 or 1·4 per cent. of carbon.

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* B. of T. tests for iron bars, 21·5 tons tensile with 25% elongation in 8 diameters.

To weld properly, steel must not contain more than ·25 per cent. of carbon. Silicon steel of quite high tensile strength will weld readily.

**Weight of steel castings.**—Soft steels, of 28 to 35 tons ultimate strength, weigh about 490 lbs. per cubic foot, or ·284 lb. per cubic inch; a plate 1 foot square and 1 inch thick will therefore weigh very nearly 41 lbs.

**The Admiralty requirements** as regards steel castings for machinery are as follows:—

Tensile strength for pistons and intricate thin castings to be 30 to 37 tons per square inch, with an extension of at least 12 per cent. in 2 inches, and for ordinary castings 28 to 35 tons per square inch, with an extension of at least 15 per cent. in 2 inches.

Bars 1 inch square to bend cold without fracture, over 1⅝ inches radius:—for ordinary castings 90° for 28-ton steel, 60° for 35-ton steel, and other strengths in proportion; and for intricate thin castings 45° independent of strength.

Test pieces are to be taken from each casting.

All steel castings are also to stand being dropped from a height of about 12 feet upon a hard road or floor.

## Lloyd's Rules for Quality and Testing of Steel Castings.

For purposes for which **cast iron** is ordinarily employed, such as propeller bosses and blades, bed-plates, engine framing and columns, brackets, weigh-shaft levers, pistons, cylinder covers, eccentric straps, bearing brushes, &c., the castings must be sound, and are to be subjected to such drop and hammering tests as below.

**Section 5.**—1. **Process of Manufacture.**—Steel for castings shall be made by the Open Hearth process, acid or basic, or by such other process as may be approved.

2. **Annealing.**—All steel castings shall be thoroughly annealed in a properly constructed annealing furnace, which must permit of the whole casting being uniformly raised in temperature throughout its whole extent to the necessary intensity. The casting shall be allowed to cool down prior to removal from the annealing furnace; and if subsequently heated for any purpose it shall again be similarly annealed if required by the Surveyor.

3. **Testing and Inspection.**—The following tests and inspections shall be made, but in the event of any casting proving unsatisfactory in the course of preparation or erection, such casting shall be rejected, notwithstanding any previous certificate of satisfactory testing.

4. **Tensile and Bend Test Pieces.**—The tensile strength and ductility shall be determined from standard test pieces, which are to be prepared from sample pieces on the casting. These sample pieces are not to be cut or partially cut from the castings until the annealing of such castings has been completed.

5. **Number of Tests.**—At least one tensile test and one cold bend test are to be taken from each casting. In castings of complex design,

referred to in paragraph 12, at least two tensile and two cold bend tests are to be taken. Where a casting is made from more than one charge of steel, at least four tensile and four cold bend tests are to be taken from pieces cast as far apart as possible on the casting, some test pieces being taken from as near the top and others from as near the bottom of the casting as practicable.

6. **Dimensions of Tensile Test Pieces.**—The tensile test pieces are to be turned to a diameter of ·564 inch with a gauge length of 2 inches, or a diameter of ·798 inch with a gauge length of 3 inches, or a diameter of ·977 inch with a gauge length of $3\frac{1}{2}$ inches.

7. **Dimensions of Bend Test Pieces.**—The bend test pieces are to be of a rectangular section, 1 inch wide by $\frac{3}{4}$ inch thick, with the edges rounded to a radius of $\frac{1}{16}$ of an inch. They are to be bent over the thinner section. The bending may be performed either by pressure or by blows.

8. **Tensile Tests.**—The tensile breaking strength determined from test pieces of standard dimensions is to be between the limits of 26 and 35 tons per square inch, with an elongation of not less than 20 per cent. measured on the standard test piece.

9. **Bend Tests.**—The bend test pieces must withstand without fracture being bent cold through an angle of 120 degrees, the internal radius of the bend being not greater than one inch.

10. **Additional Tests before Rejection.**—Should either the tensile or bend test or both fail and the Surveyor consider the fractured test piece or test pieces, or the results obtained therefrom, do not fairly represent the quality of the casting, a duplicate of the test or tests which failed shall be made if requested. In such cases the quality of the casting shall be judged by the result of the duplicate test or tests, and not by the original test or tests which failed.

11. **Percussive Tests.**—The castings are to be dropped on hard ground from a height of from 7 to 10 feet, according to the design, shape, and weight of the casting.

12. **Castings of Complex Design.**—Castings of complex design which would be liable to be deformed if submitted to the drop or percussive tests, may have this test dispensed with provided two tensile and two cold bend tests be made upon pieces taken from positions as far apart as possible on each casting; one tensile and one bend test being taken from as near the top and the others from as near the bottom of the casting as practicable.

13. **Hammering Tests.**—After being subjected to the percussive test, the casting in each case is to be subsequently slung up and well hammered with a sledge hammer not less in weight than 7 lbs., to satisfy the Surveyors that the casting is sound and without flaw. This hammering test is also to be applied to castings which may not have been submitted to a percussive test.

14. When the castings are to be used for purposes for which cast iron is ordinarily employed, they need not be submitted to tensile and bend tests, but they must be submitted to the drop and hammering tests specified in paragraphs 11 and 13.

15. **Branding.**—Every casting, after it has satisfactorily withstood the prescribed tests, shall be clearly and distinctly marked by the Society's Surveyor, indicating that the casting has complied with the Society's requirements.

## BOARD OF TRADE TESTS FOR CAST STEEL AND MALLEABLE CAST IRON.

### Testing of Cast-Steel and Malleable Cast-Iron Material.

### Cast Steel.

63. *Cast-steel stems, sternposts, rudders, propeller shaft brackets, steering quadrants, crossheads or tillers, and other important castings which are subject to considerable stress and strain.*

Tensile and bending tests are to be made from these castings in the manner described in sections 133 to 134. The tensile strength should be in accordance with the limits specified in section 133. It is desirable that the percentage of elongation and the bending angle should not be less than those specified in sections 133 and 134 respectively for castings of superior quality, but, provided there are no unusual features in the design of the castings and the scantlings are sufficient, the lower limit of elongation and the bending angle specified will not be objected to.

The Surveyor should also carefully examine each casting, and take all practicable measures to satisfy himself that it is sound and free from flaws and defects.

64. *Steel castings other than those referred to above, when fitted in positions such that the failure of the castings may affect the safety of the vessel.*

(*a*) *Side-scuttle frames and plugs.*

In cases where side-scuttle frames and plugs are made of cast-steel, the Surveyors should be guided by the following instructions as to tests.

When side-scuttles are fitted above the vessel's upper deck, or when in such a position that their sills will be at least 10 feet above the centre of the freeboard disc, no tests will be required, unless the Surveyor has reason to doubt the strength of the material. When intended to be fitted in a lower position, one frame and also one plug should be selected at random out of each fifty, and at least one frame and one plug for every vessel so fitted, and tested to destruction by bending by means of blows from a hammer when cold and before being machined.

Side-scuttle frames of ordinary form should be capable of being bent to an angle of at least 20° without fracture.

Plugs should be capable of being bent through an angle of at least 40° without fracture.

## Malleable Cast Iron.

65. Owing to the great differences in the qualities of this material as produced by different methods of manufacture, its employment in the construction of passenger steamships should be watched with care.

The surface of malleable iron castings should not be removed by machining to a greater extent than is absolutely necessary.

### 66. *Side-scuttle frames and plugs.*

Side-scuttle frames and plugs should be capable of being bent through an angle of at least 15° and 30° respectively without fracture.

### 67. *Rail Stanchions.*

Rail stanchions of malleable cast iron may be accepted for vessels intended to ply exclusively in smooth water or partially smooth water limits, provided the stanchions are capable of withstanding the following test:—

Stanchions should not be passed unless they are capable of being bent while cold at least 6 inches from the straight in a length of 36 inches, before fracture.

Malleable cast-iron rail stanchions are not sanctioned for use in sea-going vessels.

## Steel Bars and Plates.

**Mild steel** has now almost entirely superseded wrought iron for bars and plates; even for ventilators, uptakes, chimneys, casings, &c., it is cheaper than iron of the quality to stand the necessary working, rolling, &c.

In steel plates rolled from ingots the direction in which the test pieces are cut from the plates (*i.e.* in the direction of rolling or across that direction) is found to have less influence on the strengths obtained than is the case with iron plates, the differences observed usually lying between 5 and 10 per cent.

The percentage of carbon in mild steel bars and plates (27 to 30 tons per square inch) is usually between ·15 and ·25 per cent.

**Weight of steel bars and plates.**—The average weight of steel of this description is about 490 lbs. per cubic foot, or ·284 lb. per cubic inch, and a plate 1 foot square and 1 inch thick weighs about 41 lbs.

Experience in handling mild steel has shown that work cannot safely be continued after the red colour of the heat has disappeared; when cold, the material will stand very severe treatment, and it is equally ductile at a full red heat, but below 700° C. it seems to be in a critical and at times brittle and unreliable condition, and should not be handled again until it has cooled below 400° C., or been re-heated.

Also, when a plate has been locally heated or worked, the internal stresses set up seem to be much more severe in effect than in the case of wrought iron, and to avoid all risk of cracks, the plates should be carefully annealed as soon as possible.

## Table CXXXI.—Comparative Requirements for Steel Castings.

| | BOARD OF TRADE. | | | BRITISH STANDARDS COMMITTEE. | | | | LLOYD'S | BRITISH CORPORATION. | |
|---|---|---|---|---|---|---|---|---|---|---|
| Quality. | Superior. | — | — | A. | B. | C. | D. | — | — | Thin castings. |
| Tensile Strength. | 26/35 | 35/40 | 26/35 | 35/40 | 26/35 | 26/35 | No tensile or bending tests required. | 26/35 | 26/35 | 26/35 |
| Elongation. | 20 % | 15 % | 15 % | 15% | 20 % | 15 % | | 20 % | 20 % for 26 tons. 15 % for 35 tons. | May be reduced to 15% and 10% |
| Angle of Bend. | 120° | 60° | 90° | 60° | 120° | 90° | | 120° | 120° | 120° |
| Other Tests Required. | | | | | | | Hammer test only. | Percussion test required, also hammering tests. | Drop test and hammering test. | May be exempted from drop test. Hammering test to be applied. |
| Remarks. | Compares with Quality "B" of the British Standards Committee. | Compares with Quality "A" of the British Standards Committee. | Compares with Quality "C" of the British Standards Committee. | | | | No corresponding Quality in Board of Trade Requirements. | Compares with Board of Trade "Superior" and British Standards Committee Quality "B" | Elongation slightly altered from Quality "B" of British Standards Committee. | No corresponding quality in either Board of Trade, Lloyd's or British Standards Committee Requirements. |

## Steel Forgings.

Steel forgings are similar in most respects to bars and plates, but, of course, not quite so dense or so uniform in texture and strength.

**The Admiralty require** that all important steel forgings for machinery shall be made from ingots, and test pieces from all important ingots and forgings must satisfy the following conditions:—

Ultimate tensile strength to be between 30 and 35 tons per square inch, and extension not less than 27 per cent. in 2 inches.

Bars 1 inch square should bend cold, without fracture, through an angle of 180° over a radius not greater than ¼ inch.

The ultimate tensile strength of material for crank and propeller shafts is to lie between 28 and 32 tons per square inch, and the extension must be at least 30 per cent. in 2 inches. 30 per cent. of top end of ingot to be removed before forging, and at least 3 per cent. from bottom end after forging. Sectional area of body of forging is not to exceed one-sixth original sectional area of ingot.

**The weight of steel forgings** may be taken as about 487 lbs. per cubic foot, or ·282 lb. per cubic inch.

Whitworth's fluid compressed steel weighs about 495 lbs. per cubic foot.

## Lloyd's Rules respecting Ingot Steel Forgings.

**Section 6.—1. Process of Manufacture.**—Ingot steel for forgings shall be made by the Open Hearth process, acid, or basic, or by such other process as may be approved by the Committee.

The forgings must be sound. They are to be made from sound ingots, and from all important forgings such as cranks and propeller shafts, connecting-rods, piston-rods, the forgings must be gradually and uniformly forged. The sectional area of the body of the forgings (as forged) shall not exceed one-fifth of the sectional area of the original ingot, and no part of the forging (as forged) shall have more than two-thirds of the sectional area of the original ingot.

2. **Annealing.**—As for steel castings (*vide* page 441).

3. **Testing and Inspection.**—As for steel castings (*vide* page 441).

4. **Tensile and Bend Test Pieces.**—The tensile strength and ductility shall be determined from standard test pieces which are to be prepared from sample pieces cut lengthwise from the forging from a part of not less sectional dimensions than the body of the forging. Such standard test pieces shall be machined from the sample pieces without forging down, and the sample pieces shall not be detached from the forging until the annealing of such forging has been completed.

5. **Number of Tests.**—At least one tensile and one cold bend test are to be taken from each forging. Where a number of articles are cut from one forging, one tensile and one cold bend test from this whole forging will be sufficient.

6. **Dimensions of Tensile Test Pieces.**—As for steel casting.

7. **Dimensions of Bend Test Pieces.**—The bend test pieces are to be machined to a rectangular section 1 inch wide by ¾ inch thick, with the edges rounded to a radius of $\frac{1}{16}$th of an inch. They are to be bent over the thinner section. The bending may be performed either by pressure or by blows.

8. **Tensile Tests.**—The tensile breaking strength determined from test pieces of standard dimensions is to be between the limits of 28 and 32 tons per square inch, with an elongation on the standard test piece of not less than 29 per cent. for 28-ton steel, and 25 per cent. for 32-ton steel, and in no case must the sum of the tensile breaking strength and corresponding elongation per cent. be less than 57.

9. **Bend Tests.**—The bend test pieces must withstand without fracture being bent cold through an angle of 180 degrees, the internal radius of the bend being not greater than ¼ inch.

10. **Additional Tests before Rejection.**—Should either the tensile or bend test, or both, fail, and the Surveyor consider the fractured test piece or test pieces, or the results obtained therefrom, do not fairly represent the quality of the forging, a duplicate of the test or tests which failed shall be made if requested by the maker. In such cases the quality of the forging shall be judged by the result of the duplicate test or tests and not by the original test or tests which failed.

11. **Branding.**—Every forging, after it has satisfactorily withstood the prescribed tests, shall be clearly and distinctly marked by the Society's Surveyor, indicating that the forging has complied with the Society's requirements.

12. **General.**—The requirements as to annealing and testing are intended to apply to shafts of all descriptions, also to connecting-rods and piston rods which require to be made in several heats. They are not intended to apply to small forgings which during their last stage of manufacture are uniformly heated throughout.

## British Corporation Rules for Steel.

**The British Corporation** require the following:—

(i) Boiler shell plates and butt straps 28 to 32 tons tensile, with at least 20 per cent. elongation in 8 inches.
(ii) Plates for flanging 26 to 30 tons and 23 per cent. at least.
(iii) Plates and stays for furnaces and combustion chambers 26 to 30 tons and 23 per cent.
(iv) Rivet bars, same tensile but with 25 per cent. minimum extension or 31 per cent. with 3½ diameters.

## Copper.

Copper in its unalloyed condition is hardly ever used. For pipes and for fire-box plates of loco type boilers 0·25 to 0·45 per cent. of arsenic is added.

The tenacity of such sheet copper is about 30,000 lbs. per sq. inch.

Annealed copper wire has a strength of 40,000 lbs. per sq. inch.

Wire of copper with 1 per cent. aluminium will stand 78,000 lbs.

per square inch, and French wire having a small addition of silicon to the copper as much as 128,000 lbs. Wire of copper with 0·529 per cent. of antimony stands 78,000 lbs.

Sheet copper 8·805 specific gravity has a weight equivalent to about 550 lbs. per cubic foot, or ·318 lb. per cubic inch. A square foot 1 inch thick weighs 45·83 lbs.

For weights of copper pipes see page 492.

**The Admiralty specify** that strips cut from steam and other pipes, either longitudinally or transversely, are to have an ultimate tensile strength of not less than 13 tons when annealed in water, and are to elongate at least 35 per cent. in 2 inches, or 30 per cent. in 4 inches.* They are also to bend through 180° until the two sides touch, and to stand hammering to a fine edge, when cold, without cracking.

Copper with 2 per cent. aluminium must stand 15 tons (33,600 lbs.) per square inch and show an extension of 40 per cent. This alloy weighs about 540 lbs. per cubic foot. An alloy of 92·5 copper and 7·35 aluminium has a very high resistance to alternating stresses, is tough and has high tensile strength.

## Common Bronze or Gun-metal.†

**Bronze** is composed of copper and tin in various proportions, and a small percentage of zinc is usually added to ensure sound castings and permit of "tooling" easily.

Its strength depends mainly on the proportions and quality of the metals forming its composition, but is much affected by such circumstances as the size of the casting, the rate of cooling, and the skill of the founder in the heat treatment and mixing them, ventilating the moulds, relieving the cores, &c.

In large castings which cool slowly there is a great tendency for the metals to separate from one another to some extent, and the average strength is therefore usually less than in small castings; as a general rule the more quickly the casting is cooled the stronger the metal is.

The metal sets very rapidly, and contracts nearly $^3/_{16}$ inch per foot on an average, so that in large castings the cores must be very quickly relieved if the casting is not to be drawn and porous and of low tenacity.

With a mixture of 90 per cent. copper and 10 per cent. tin, a carefully made test bar may be got to show a strength of nearly 17 tons (38,000 lbs.) per square inch. With 84 per cent. copper and 16 per cent. tin, a much harder metal is obtained (the hardness of gun-metal varies almost directly with the percentage of tin in the mixture), with an ultimate strength of test piece of about 16 tons (35,000 lbs.) per square inch.

For heavy bearings, 79 per cent. copper and 21 per cent. tin is sometimes used; the resulting metal is very hard, and test pieces show a strength of 13½ to 14 tons (30,000 to 31,000 lbs.) per square inch.

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* B.M.E.D. & C.C. require 14 tons with elongation of 40 per cent. in 2 inches.

† Guns are no longer made of bronze, but now invariably of steel.

For bearings the metal is improved by the addition of a small amount of lead; a little zinc is always necessary to facilitate machining.

**Admiralty Bronze.**—For all ordinary castings and steam fittings in connection with the machinery, the alloy used must contain *not less* than 10 per cent. of tin, and *not more* than 2 per cent. of zinc. For air-compressing machinery and torpedo fittings, &c. (where the working pressure is over 3000 lbs. per square inch), the mixture specified is:—Copper, *not less* than 86 per cent.; tin, *not less than* 10 *nor more than* 12 per cent.; zinc, *not more* than 2 per cent.

The ultimate tensile strength of bronze is to be not less than 14 tons (31,360 lbs.) per square inch, and it must extend at least 7½ per cent. in 2 inches before breaking.

Fairly good ordinary bronze ought to show a strength of about 12 tons (27,000 lbs.) per square inch, and should extend 10 per cent. in a length of 2 inches before breaking.

**Its weight** is about 545 lbs. per cubic foot, or ·315 lb. per cubic inch.

## Phosphor Bronze.

This metal is composed of copper and tin, with a small proportion (about ½ per cent.) of phosphorus. It is harder than ordinary gun-metal, very close-grained, and of superior strength. The average ultimate strength is about 15½ tons (35,000 lbs.) per square inch, while that of some grades of the metal is as high as 22 tons; it is, however, "red-short," so that when heated it is liable to crack. Great care is required in melting and running it, and repeated meltings very much reduce its virtue, as the phosphorus disappears. Sheet phosphor bronze and rods, wire, &c., are most useful, and this metal is now often used for the blades of turbines. The elastic limit of phosphor bronze is very high.

## Manganese Bronze.

Manganese bronze now consists of copper, tin, and zinc (according to the grade of metal required), with the addition of a proportion of ferro-manganese.

**The weight** of this material is about 535 lbs. per cubic foot, or ·31 lb. per cubic inch.

Rolled rods of manganese bronze can be obtained of strengths varying from 28 to 32 tons (63,000 to 72,000 lbs.) per square inch, and showing an elongation of 40 per cent. to 15 per cent in 2 inches.

## Zinc Bronzes.

**Parsons' Manganese Bronze Co.** were the first to manufacture manganese bronze for propellers, &c.; their alloy formerly had a very considerable content of manganese and possessed a high tensile strength, &c. It has of late years been replaced by alloys having much less manganese, but some other metals in small quantities, which, with the methods of melting and treating followed by the company, possess quite as high and often higher strength, with greater toughness, and safer for marine castings. The following are the chief products of this company:—

*Crotarite*, suitable for boiler stays and other fittings exposed to heat. The melting point is nearly as high as copper (2000° F.), it is highly malleable, and has a tensile strength of 25 to 26 tons, with 30 to 40 per cent. elongation. Elastic limit is 16 tons.

*Immadium*, suitable for shafts, pumps, rods, &c., exposed to sea-water, being incorrodible. Mild variety has an elastic limit of 18 tons and an ultimate tensile of 36 tons, with 25 per cent. elongation. In the cast state it is very suitable for boiler mountings and fittings, as it has an elastic limit of 9 tons at 500° F. and an ultimate of 23·5 tons, with 22·5 per cent. elongation.

*Turbadium* is a special bronze for high-speed propellers, being capable of resisting the corrosion of sea-water and erosion due to high velocity. The elastic limit is 18 tons and the ultimate tensile 40 tons, with an extension of 18·5 per cent. in 8 inches. This can be wrought at a cherry-red heat.

*White Brass*, as a bearing metal, is another product of this company.

**J. Stone & Co.'s Bronzes.**—This firm supply several kinds of copper alloys suitable for the different parts of marine machinery, the principal of which are:—

1. *Toughened Copper*, suitable for boiler stays and other fittings which require to be tough and strong when at a fairly high temperature. At 60° F. its ultimate tensile is 18·1 tons per square inch, with an elongation of 55 per cent. in 2 inches; while at 400° F. it is still 55 per cent., with 14·1 tons ultimate and 7·4 tons elastic limit. At 800° F. the elastic limit is 8·7 tons, with an elongation at fracture of 37 per cent.

2. *Special Manganese Bronze*, for propellers of high revolution, is guaranteed to have an ultimate tensile of 32 tons, with an elongation of 20 per cent. in 2 inches.

3. *Patent Bronze* (*No.* 4 *quality*) when in cold rolled bars have an ultimate strength of 45 tons, with 10 per cent. elongation in 2 inches and an elastic limit of 37·5 tons.

In the annealed state the ultimate is 34 tons and 36 per cent., with the elastic limit at 17 tons.

In normal state it withstands 40 tons per square inch compression, with a shortening of only ·084 per inch, with a bulge of ·035 in a diameter of ¾ inch.

4. *Stone's Ordinary Propeller Bronze.*—Six Admiralty tests gave an average of 33·8 tons ultimate tensile, with an elongation of 23·5 per cent. in 2 inches.

5. *White Bronze* for bearings, &c., is composed largely of tin.

**Bull's Metal Company** manufacture another of the zinc bronzes for propellers, which has an ultimate tenacity of 31 tons, with an elongation of 25 per cent.; its elastic limit is 13·2 tons. When rolled the tensile is 34 tons, with an elongation of 34 per cent. in 2 inches and an elastic limit of 26·8 tons. When this metal is at a temperature of 400° F., the ultimate elongation is 14·6 per cent., while the elastic limit is 21 tons.

*Melloid.*—A special alloy supplied by this company to resist heat, has a tensile of 98,000 at 60° F., while at 600° F. it is as high as 79,000 lbs. or 35 tons, with an elongation of 18 per cent. in 2 inches; and even at 800° F. it has an ultimate tenacity of 31,800 lbs., with an elongation of 50 per cent.

## Brass.

**Brass** is essentially an alloy of copper and zinc only, of which there are many variations in composition. The most important in marine engineering are:—

1. *Muntz Metal*, an alloy of 59 to 60 per cent. of copper with 41 to 40 of zinc. It should have a tensile strength of at least 22 tons, with an elongation of 27 per cent.; when rolled it has an elastic limit of 36,000 lbs. It is very ductile and can be forged hot. Weight 512 lbs. per cubic foot, 0·296 per cubic inch.

2. *Condenser Tube Metal* is usually made with 70 per cent. copper and 30 zinc, and should in the drawn state have a tensile strength of 36 tons ultimate. Admiralty condenser tubes are made of 70 per cent. best selected copper, 29 of silesian zinc, and 1 of tin.

3. Condenser tubes of a mixture of 70 copper, 28 zinc, and 2 tin are said to resist corrosion even better than the Admiralty mixture.*

4. *Naval Brass* is virtually Muntz metal with an addition of 1 per cent. of tin. It is generally composed of 62 copper, 31 zinc, and 1 tin; and in the wrought state should have an ultimate tensile strength of 60,800 lbs., with an elongation of 19 per cent., and the elastic limit should be 42,400 lbs. It can be forged, &c., and withstands the action of sea-water better than Muntz metal.

The Admiralty require naval brass rods of ¾ inch and upwards to have an ultimate tensile of 22 tons and an extension *before* fracture of at least 10 per cent. in 2 inches; they must also bend through 75° with a radius of one diameter; and when hot are to be forged to a fine point.

*Yellow Brass*, for ornamental and ordinary castings where strength is not wanted, is composed of 2 copper to 1 zinc. The addition of a little lead improves the colour and facilitates machining. Its ultimate tensile is about 11 tons.

**Aluminium** is now very extensively used in the arts, and from its lightness would be a most useful one for marine engineers if it did not become so quickly and seriously affected by sea-water. In the pure state it is practically of no use to them, but when alloyed its tensile strength and durability are much improved. It is now so cheap that it can be used for any purpose so far as cost is concerned, and is used for telegraph wire in place of copper in certain localities, especially since copper has been so dear (£72 per ton). Its conductivity is good; its resistance is 2·839 per c.c. as against 1·621 copper wire and 9·637 iron. Its ultimate tensile in wire form is 8 tons. When alloyed with

---

* The Muntz Metal Co. substitute lead for tin with very satisfactory results.

6 per cent. of copper and rolled into sheets, rods, &c., it has a strength of 11 to 12 tons after, and 14 to 16 tons before annealing.

**Duralumin** is a 90 per cent. alloy of aluminium made by Vickers Limited, having a specific gravity of only 2·8, a melting point of 1200° F., an ultimate tensile as high as 40 tons, when the extension, however, is small; at 28 to 30 tons ultimate, the extension is 15 per cent. in 2 inches, while with 25 tons it is 20 per cent.

**Monel Metal** is a natural nickel bronze, having a very high tensile strength both in the cast and rolled state. Castings have an ultimate tensile from 33·7 to 36·5 tons, with an elongation of 38 to 28·5 per cent.; the elastic limit being 16 to 18 tons per square inch in 2 inches.

Rolled bars, plates, &c., have an elastic limit of 25 tons and an ultimate of 42 tons, with an elongation of 41 per cent. Can be forged hot and brazed. This alloy is uncorrodible and used for screw propellers of high-speed craft in the U.S.A. Navy. Its specific gravity is 8·87; weight per cubic inch 0·319 lbs. cast, 0·323 rolled; melting point 2480° F.

**The Delta Metal Co.** produces several varieties of zinc bronze and other alloys.

**D. M.** No. 1, of fine golden colour, very hard, resists corrosion, and takes place of steel. As cast, it has an ultimate strength of nearly 41 tons, with an elongation of 20 per cent. At a red heat it is highly malleable. When wrought, the ultimate tensile is 43·27 tons, with an elongation of 27 per cent.

This metal is used for extrusion into bars of various sections, the ultimate strength of which is nearly 50 tons, with 26 per cent. elongation and 24·5 tons yield point.

It has high resistance to corrosion from action of sea water and acids. A shearing test showed its resistance to be no less than 23·11 tons per square inch.

**D. M.** No. 2 is a silver white metal, malleable, takes a high polish, and therefore suitable for fittings, &c. When extruded it has a yield point of 22 tons, and the ultimate strength 42·45 tons with 14 per cent. of elongation. As cast, it has an ultimate of 38 tons with 10 per cent.

**D. M.** No. 3 is suitable for condenser tubes and pipes generally, as it can be easily drawn and resists corrosion.

**D. M.** No. 4 is equal to steel in strength and toughness, strongly resists corrosive effects of sea water and acids, chemical gases, &c., therefore largely used in chemical engineering as well as marine. It is used also in the cast form, and bars cut from chill castings gave 19 tons limit of elasticity, 26·8 ultimate with an elongation of 36·8 per cent.

It can be forged or drawn into tubes when the ultimate is 34·4 tons with 26·25 per cent. elongation. It can also be stamped with equally good results.

It is often used in sheets for boat-building and similar purposes, and the company supplies for this purpose sectional bars, angles, zeds, channels, &c. &c.

**D. M.** No. 5 is a bearing metal.

**D. M.** No. 7 is suitable for high temperature.

## Table CXXXII.—Composition of White (Bearing) Metals.

| Name of Metal. | Cu. | Sn. | Sb. | Zn. | Pb. | Fe. | Miscellaneous. |
|---|---|---|---|---|---|---|---|
| Dewrance's Locomotive, | 22·2 | 33·3 | 44·4 | ... | ... | ... | ... |
| For gland packings, . | 7·8 | 88·1 | 3·5 | ... | 2·6 | ... | ... |
| Used in the German Navy, . . . | 7·5 | 85·0 | 7·5 | ... | ... | ... | ... |
| ,, French Navy, | 7·0 | 7·5 | ... | 78·5 | 7·0 | ... | ... |
| ,, British Navy, | 5·5 | 86·0 | 8·5 | ... | ... | ... | ... |
| Babbit's Metal, . . | 8·5 | 83·0 | 8·5 | ... | ... | ... | ... |
| Fenton's ,, . . | 4·4 | 16·6 | ... | 79·0 | ... | ... | ... |
| Magnolia, ,, . . | ... | ... | 21·0 | ... | 78·0 | 1·0 | ... |
| ,, ,, . . | ... | 4·6 | 13·0 | ... | 82·0 | ... | 0·4 |
| Kingston's Metal, . | 6·0 | 88·0 | ... | ... | ... | ... | 6 Hg |
| Parson's white brass, . | 5·6 | 17·5 | 0·8 | 76·1 | ... | ... | ... |
| ,, ,, metal, . | ... | 58·5 | 2·0 | 39·5 | ... | ... | ... |
| ,, ,, ,, . | 1·0 | 68·0 | ... | 30·5 | 0·5 | ... | ... |
| For common bearings, . | 10·0 | ... | 10·0 | ... | 80·0 | ... | ... |
| ,, heavily loaded bearings, . . . | 64·0 | 5·0 | ... | ... | 30·0 | ... | 1 Ni |
| Plumtine, . . . | 0·5 | 40·5 | 11·0 | ... | 48·5 | ... | ... |

A specially good white metal for heavy bearings is made by mixing 6 parts of tin with 1 of copper, and 6 parts of tin with 1 of antimony, and then adding the two mixtures together.

The exact Admiralty specification is,—at least 85 per cent. tin, not less than 8 per cent. antimony, and about 5 per cent. copper ; zinc or lead not to be used. This is somewhat harder than the above.

## Table CXXXIII.—Properties of Metals.

| Metal or Alloy. | Ultimate Strength. | Extension in 8 inches. | Elastic Limit. | Compressed Strength. | Shear per cent. of Tensile. | Specific Gravity. | Weight of | | Melts at ° F. | Conducts Heat. Silver 100 | Extension per 10° F. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Cubic Inch. | Sq. Ft. 1 inch Thick. | | | |
| | Lbs. | Per cent. | Lbs. | Lbs. | | | | | | | |
| Cast iron, good castings, | 25,000 | ... | ... | 110,000 | 162·0 | 7·20 | 0·260 | 37·50 | 2100 | ... | ... |
| Wrought iron, good boiler quality, | 50,000 | 12·0 | 26,000 | ... | 80·0 | 7·68 | 0·280 | 40·0 | 2800 | 11·90 | ·0000670 |
| ,, ,, bar ,, | 60,000 | 15·0 | 30,000 | ... | ... | 7·62 | 0·276 | 39·70 | ,, | ,, | ,, |
| ,, ship plate ,, | 45,000 | 10·0 | 27,500 | ... | ... | 7·68 | 0·280 | 40·0 | ,, | ,, | ,, |
| Steel castings, good mild, | 63,000 | 15·0 | 34,000 | ... | ... | ... | ... | .. | 2500 | ,, | ,, |
| ,, wrought (0·15 carbon), | 58,000 | 25·0 | 29,000 | ... | 80·0 | 7·86 | 0·284 | 40·94 | ,, | ,, | ,, |
| ,, ,, (0·40 ,, ), | 78,000 | 22·0 | 33,000 | ... | ... | ... | ... | .. | ... | ... | .. |
| ,, ,, ( ,, ,, ), oil tempered, | 90,000 | ... | 55,000 | ... | ... | .. | ... | ... | ... | ... | ... |
| Nickel steel (3½ Ni, 0·29 C), forged, | 85,000 | 30·0 | 54,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| ,, (3¾ Ni), oil tempered, | 90,000 | ... | 60,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| ,, mild quality boiler plate, | 74,000 | ... | 43,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| ,, high tensile ,, | 114,000 | ... | 64,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| Chrome steel (2·5 Cr, 0·39 C), forged, | 65,400 | ... | 52,600 | ... | ... | ... | .. | ... | ... | ... | ... |
| ,, ( ,, ), unannealed, | 120,900 | ... | 97,400 | ... | ... | ... | ... | ... | ... | ... | ... |
| Vanadium steel (0·257, V), | 88,000 | 24·0 | 76,300 | ... | ... | ... | ... | ... | ... | ... | ... |
| Chrome vanadium steel (1·0 Cr + ·15 V.), | 117,800 | 25·0 | 77,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| Manganese steel (11·0 Mn), forged, | 134,000 | 35·0 | ... | .. | ... | ... | ... | ... | ... | ... | ... |
| Nickel, cast, | 36,400 | ... | 24,600 | ... | ... | 8·30 | ... | ... | 2820 | ... | ... |
| ,, forged, | 72,100 | ... | 25,500 | ... | ... | 8·80 | 0·318 | 45·80 | 2820 | ... | ·0000705 |
| German silver (19 Ni), rolled, | ... | ... | ... | ... | ... | 8·80 | 0·318 | 45·80 | ... | ... | ... |
| Copper, annealed sheets, | 30,000 | ... | 5,600 | ... | 86·7 | 8·85 | 0·320 | 46·10 | 2000 | 73·6 | ·0000930 |
| ,, bars | 36,000 | ... | ... | ... | ... | ,, | ... | ... | ,, | ,, | ,, |
| Zinc, cast, | 3,000 | ... | ... | .. | ... | 7·00 | 0·253 | 36·42 | 800 | 28·1 | ·0001620 |
| ,, rolled, | 7,500 | ... | ... | ... | ... | ,, | ... | ... | ,, | ,, | ,, |
| Tin, cast, | 4,700 | ... | ... | ... | ... | 7·30 | 0·264 | 38·00 | 440 | 15·2 | ·0001280 |
| Muntz metal, rolled, | 56,000 | 27·0 | 36,000 | ... | .. | 8·20 | 0·296 | 42·72 | 1630 | ... | ... |
| Naval brass | 60,800 | 19·0 | 42,400 | ... | 55·8 | 8·20 | 0·296 | 42·72 | 1640 | ... | ... |
| Cobalt (cast annealed) | 36,000 | | | 12,200 | | 8·81 | | | | | |

## Table CXXXIII.—Properties of Metals—*continued.*

| Metal or Alloy. | Ultimate Strength. | Extension in 2 inches. | Elastic Limit. | Compressed Strength. | Shear per cent. of Tensile. | Specific Gravity. | Weight of | | Melts at ° F. | Conducts Heat. Silver 100 | Extension per 10° F. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | Cubic Inch. | Sq. Ft. 1 inch Thick. | | | |
| | Lbs. | Per cent. | Lbs. | Lbs. | | | | | | | |
| Manganese bronze (Parsons), cast No. 2, | 67,000 | 24·5 | 33,000 | ... | ... | 8·51 | 0·308 | 44·34 | 1660 | ... | ... |
| ,, ,, rolled, annealed, | 67,500 | 26·2 | 33,000 | ... | ... | 8·40 | 0·304 | 43·76 | ,, | ... | .. |
| Stone's bronze, cast, | 77,500 | ... | 38,200 | 127,000 | ... | .. | ... | ... | ... | ... | ... |
| ,, rolled, | 80,600 | .. | 66,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| Delta metal, rolled, | 89,000 | ... | ~ | ... | ... | 8·40 | 0·304 | 43·76 | 1800 | ... | ... |
| Bull's metal, cast, | 70,000 | ... | 29,600 | 176,000 | ... | 8·30 | 0·300 | 43·24 | 1800 | ... | ... |
| ,, rolled, | 78,000 | 34·0 | 60,000 | 224,000 | ... | ,, | ,, | ,, | ,, | ... | ... |
| Aich's metal, ,, | 57,300 | 25·6 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| Phosphor bronze (wheel B), cast, | 39,200 | ... | 38,900 | ... | 113·0 | 8·60 | 0·311 | 44·80 | 1900 | ... | ... |
| ,, ,, rolled, | 68,500 | ... | 62,000 | ... | 63·0 | ,, | ,, | ,, | ,, | ... | ... |
| Silicium bronze wire, | 105,000 | ... | ... | ... | ... | ... | ... | .. | ... | ... | ... |
| Admiralty bronze, cast | 30,000 | ... | 17,000 | ... | ... | 8·66 | 0·325 | 46·80 | 1700 | ... | ... |
| Aluminium (pure comml.), cast, | 11,800 | ... | ... | ... | ... | 2·56 | 0·092 | 13·33 | 1160 | 31·33 | ·0000280 |
| ,, ( ,, ), bar, rolled, | 28,200 | ... | 19,300 | ... | ... | 2·70 | 0·098 | 14·06 | ,, | ,, | ,, |
| ,, ( ,, ), plates, rolled, | 16,500 | ... | 9,600 | ... | ... | 2·70 | 0·098 | 14·06 | ,, | ,, | ,, |
| ,, (6·0 Cu), plates rolled, | 34,400 | 3·7 | 30,600 | ... | ... | 3·07 | 0·111 | 16·00 | ... | ... | ... |
| Aluminium bronze (10 Al), rolled | 100,000 | 22·0 | ... | ... | ... | 7·65 | 0·277 | 39·86 | 1750 | ... | ... |
| ,, (5 Al), cast, | 56,000 | 50·0 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| Antimony, cast, | ... | ... | ... | .. | ... | 6·71 | 0·243 | 35·00 | 1166 | 4·03 | ... |
| Bismuth, ,, | ... | ... | ... | ... | ... | 9·80 | 0·355 | 51·05 | 514 | 1·80 | .. |
| Lead, sheet, | 3,300 | ... | ... | ... | ... | 11·37 | 0·411 | 59·24 | 620 | 8·50 | ... |
| Manganese, cast, | ... | ... | ... | ... | ... | 8·01 | 0·289 | 41·52 | 3450 | ... | ... |
| Platinum, rolled, | ... | ... | ... | ... | ... | 21·50 | 0·778 | 112·0 | 3225 | 8·40 | ... |
| Silver, rolled, | ... | ... | .. | ... | ... | 10·53 | 0·381 | 54·86 | 1730 | 100·0 | ... |
| Parsons' immadium, cast, | 80,600 | 15·0 | 36,100 | ... | ... | ... | ... | ... | ... | ... | ... |
| ,, ,, forged, | 90,000 | 20·0 | 45,000 | ... | ... | ... | ... | ... | ... | ... | ... |
| ,, turbadium, cast, | 90,200 | 18·58 | 40,320 | ... | ... | ... | ... | ... | ... | ... | ... |
| Monel metal, cast, | 78,500 | 33·0 | 38,100 | ... | ... | 8·87 | 0·319 | 46·0 | 2480 | 17·0 | ·000096 |

## Table CXXXIV.—Prices of Materials.*

| Materials at Maker's Works. | Prices, 1914. | August 1911. |
|---|---|---|
| Pig iron, Shropshire, cold blast, No. 3, | £6 5 0 ⅌ ton. | £5 5 0 |
| Pig iron, Hæmatite, West Coast, No. 3, | 3 5 0 ,, | 3 8 0 |
| Pig iron, Hæmatite, East Coast, No. 3, | 3 3 0 ,, | 3 3 0 |
| Pig iron, Scotch, good brand, No. 3, | 2 16 0 ,, | 2 18 0 |
| Pig iron, Scotch, good brand, No. 1, | 3 10 0 ,, | 3 3 0 |
| Pig iron, Cleveland, good brand, No. 3, | 2 10 0 ,, | 2 7 0 |
| Pig iron, Cleveland, good brand, No. 1, | 2 12 6 ,, | 2 10 6 |
| Pig iron, Lincoln, good brand, Foundry, | 2 13 0 ,, | 2 8 6 |
| Pig iron, Derby, Leicester, good brand, Foundry, | 2 18 0 ,, | 2 10 0 |
| Wrought iron, Cleveland bars, ordinary, | 7 10 0 ,, | 6 10 0 |
| Wrought iron, Cleveland bars, best, | 7 17 6 ,, | 7 0 0 |
| Wrought iron, Cleveland ship plates, | 6 15 0 ,, | 6 15 0 |
| Wrought iron, Cleveland ship angles, | 7 10 0 ,, | 6 15 0 |
| Wrought iron, Staffordshire best bar, | 7 0 0 ,, | 6 17 6 |
| Wrought iron sheets, | 9 5 0 ,, | 7 12 6 |
| Steel ,, ,, | 8 5 0 ,, | |
| ,, angles and tees, ship quality, | 6 2 6 ,, | 6 7 6 |
| Steel plates, ship quality, | 6 10 0 ,, | 6 15 0 |
| ,, boiler plates, basis, | 7 15 0 ,, | 7 10 0 |
| ,, forgings, plain shafts with couplings, | £20 to 25 0 0 ,, | £10 to £15 |
| Steel for tools, common, | 4d. to 6d. ⅌ lb. | 4d. to 6d. |
| ,, for tools, good, | 6d. to 9d. ,, | 6d. to 9d. |
| ,, for tools, self-hardening, for high-speed, | £0 1 6 ,, | £0 1 3 |
| Steel for tools, very best, for high speed, | 0 2 0 ,, | 2s. 3d. to 2s. 6d. |
| Copper ingots, G.M.B., | 65 5 0 ⅌ ton. | £56 12 6 ⅌ ton. |
| ,, ,, tough, | 71 5 0 ,, | 60 5 0 ,, |

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* Now altogether abnormal.

## Table CXXXIV.—Prices of Materials—(*continued*).

| Materials at Maker's Works. | Prices, 1914. | | | | August 1911. | | |
|---|---|---|---|---|---|---|---|
| Copper ingots, best selected, | £71 | 5 | 0 | ℔ ton. | £60 | 5 | 0 |
| ,, locomotive plates, | 83 | 0 | 0 | ,, | 73 | 0 | 0 |
| ,, sheets, basis, | 81 | 0 | 0 | ,, | 71 | 0 | 0 |
| ,, pipes, solid drawn, 1¼ in. to 3 in., | 0 | 0 | 10¼ | ℔ lb. | 0 | 0 | 8½ |
| Copper pipes, solid drawn, under 1¼ in. and over 3 in., | 0 | 0 | 10¼ | ,, | 0 | 0 | 8¾ |
| Copper pipes, brazed, | 0 | 0 | 8¾ | ,, | 0 | 0 | 8¼ |
| Brass condenser plates, Mercantile, | 0 | 0 | 7½ | ,, | 0 | 0 | 6¼ |
| Brass condenser plates, Naval, | 0 | 0 | 8 | ,, | 0 | 0 | 6¾ |
| Brass tubes, 66% copper, | 0 | 0 | 8⅜ | ,, | 0 | 0 | 6¾ |
| Brass condenser tubes, 70% copper, Mercantile, | 0 | 0 | 8⅜ | ,, | 0 | 0 | 6¾ |
| Brass condenser tubes, 70% copper, Admiralty, | 0 | 0 | 8⅞ | ,, | 0 | 0 | 7¼ |
| Tin English ingots, | 170 | 10 | 0 | ℔ ton. | 190 | 0 | 0 |
| Zinc slabs, Silesian, | 21 | 11 | 0 | ,, | 26 | 10 | 0 |
| ,, ,, best, | 22 | 7 | 6 | ,, | 27 | 5 | 0 |
| Aluminium ingots, Nos. 1, 2, and 4, | 95 | 0 | 0 | ,, | | | |
| Aluminium ingots, No. 6, | 88 | 0 | 0 | ,, | 61 | 0 | 0 |
| ,, sheets, from Nos. 1 to 30, wire gauge, | £130 to 150 | 0 | 0 | ,, | | | |
| Lead pig, | 18 | 10 | 0 | ,, | 14 | 0 | 0 |
| ,, pipe, | £14 10s. to 15 | 0 | 0 | ,, | 16 | 2 | 6 |
| Nickel ingots, of good q'l'ty, | 0 | 1 | 6½ | ℔ lb. | | | |
| Muntz metal sheet, | 0 | 0 | 8½ | ,, | 0 | 0 | 6¼ |
| Naval brass rods, | 0 | 0 | 8¾ | ,, | 0 | 0 | 6½ |
| Phosphor bronze ingots, for general castings, | 98 | 0 | 0 | ℔ ton. | £88 to £92 (basis). | | |
| Phosphor bronze ingots, suitable for bearings, | 104 | 0 | 0 | ,, | | | |
| Phosphor bronze rods, | 0 | 1 | 1½ | ℔ lb. | | 0 | 11¾ |
| ,, sheet, | 0 | 1 | 2 | ,, | 0 | 1 | 0½ |
| Manganese bronze ingots, | £70 to 80 | 0 | 0 | ℔ ton. | | | |
| Antimony cakes, | 28 | 0 | 0 | ,, | £28 | 2 | 6 |
| Bismuth, | 0 | 8 | 3 | ℔ lb. | | | |
| Platinum wire, | 5 | 5 | 0 | ℔ oz. troy. | | | |
| ,, sheets, | 5 | 12 | 0 | ,, | | | |
| Copper wire, | 0 | 0 | 9 | ℔ lb. | | | |
| Steel ,, | 0 | 0 | 1 | ,, | | | |
| Aluminium wire, | 0 | 1 | 0 | ,, | | | |

# PLATES OBTAINABLE IN GREAT BRITAIN.

## Area List for Sheared Steel Plates.

Maximum dimensions of plates rolled, *except by special arrangement.*

| Rectangular Plates. Maximum. | | | | | | Circular Plates. Maximum. | | |
|---|---|---|---|---|---|---|---|---|
| Thickness. | Length. | | Width. | | Area in feet. | Thickness. | Diameter in feet. | |
| ins. | ft. | ins. | ft. | ins. | ft. | ins. | ft. | ins. |
| $\frac{1}{4}$ | 30 | 0 | 6 | 0 | 120 | $\frac{1}{4}$ | 6 | 6 |
| $\frac{5}{16}$ | 30 | 0 | 6 | 0 | 132 | $\frac{5}{16}$ | 7 | 0 |
| $\frac{3}{8}$ | 35 | 0 | 8 | 0 | 190 | $\frac{3}{8}$ | 9 | 0 |
| $\frac{7}{16}$ | 40 | 0 | 8 | 0 | 200 | $\frac{7}{16}$ | 9 | 6 |
| $\frac{1}{2}$ | 40 | 0 | 10 | 0 | 240 | $\frac{1}{2}$ | 11 | 0 |
| $\frac{9}{16}$ | 50 | 0 | 10 | 0 | 260 | $\frac{9}{16}$ | 11 | 0 |
| $\frac{5}{8}$ | 50 | 0 | 11 | 0 | 300 | $\frac{5}{8}$ | 12 | 0 |
| $\frac{3}{4}$ | 50 | 0 | 12 | 6 | 325 | $\frac{3}{4}$ | 13 | 0 |
| $\frac{7}{8}$ | 50 | 0 | 12 | 6 | 350 | $\frac{7}{8}$ | 13 | 0 |
| 1 | 50 | 0 | 12 | 6 | 430 | 1 | 13 | 0 |
| $1\frac{1}{8}$ | 50 | 0 | 12 | 6 | 450 | $1\frac{1}{8}$ | 13 | 0 |
| $1\frac{3}{16}$ | 50 | 0 | 12 | 6 | 430 | $1\frac{3}{16}$ | 13 | 0 |
| $1\frac{1}{4}$ | 50 | 0 | 12 | 6 | 415 | $1\frac{1}{4}$ | 13 | 0 |
| $1\frac{3}{8}$ | 50 | 0 | 12 | 6 | 380 | $1\frac{3}{8}$ | 13 | 0 |
| $1\frac{1}{2}$ | 50 | 0 | 12 | 6 | 340 | $1\frac{1}{2}$ | 13 | 0 |
| $1\frac{5}{8}$ | 50 | 0 | 12 | 6 | 300 | $1\frac{5}{8}$ | 13 | 0 |

## Area List for Thick Plates, which can be Planed to Sizes.

| Thickness. | Maximum Length. | | Maximum Width. | | Maximum Area. |
|---|---|---|---|---|---|
| ins. | ft. | ins. | ft. | ins. | ft. |
| $1\frac{3}{4}$ | 30 | 0 | 12 | 0 | 288 |
| 2 | 25 | 0 | 12 | 0 | 250 |
| $2\frac{1}{2}$ | 25 | 0 | 12 | 0 | 200 |
| 3 | 25 | 0 | 12 | 0 | 160 |
| 4 | 25 | 0 | 11 | 0 | 120 |
| 5 | 20 | 0 | 10 | 0 | 100 |
| 6 | 20 | 0 | 9 | 0 | 81 |

The area divided by the length gives the width of a plate that can be rolled in any given thickness, and the area divided by the width gives the length.

# STEEL BARS OBTAINABLE IN GREAT BRITAIN.

## Section List of Bars.

### FLATS.

| Width. ins. | | Thickness. in. | | ins. | Width. ins. | | Thickness. in. | | ins. | Width. ins. | | Thickness. ins. | | ins. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $\frac{1}{2}$ | × | $\frac{1}{8}$ | to | $\frac{3}{8}$ | 2 | × | $\frac{1}{4}$ | to | $1\frac{3}{4}$ | 5 | × | $\frac{3}{8}$ | to | 4 |
| $\frac{5}{8}$ | × | $\frac{1}{8}$ | ,, | $\frac{1}{2}$ | $2\frac{1}{4}$ | × | $\frac{1}{4}$ | ,, | 2 | $5\frac{1}{4}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $\frac{11}{16}$ | × | $\frac{1}{8}$ | ,, | $\frac{5}{8}$ | $2\frac{3}{8}$ | × | $\frac{1}{4}$ | ,, | 2 | $5\frac{1}{2}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $\frac{3}{4}$ | × | $\frac{1}{8}$ | ,, | $\frac{5}{8}$ | $2\frac{1}{2}$ | × | $\frac{1}{4}$ | ,, | $2\frac{1}{4}$ | $5\frac{3}{4}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $\frac{13}{16}$ | × | $\frac{1}{8}$ | ,, | $\frac{5}{8}$ | $2\frac{3}{4}$ | × | $\frac{1}{4}$ | ,, | $2\frac{1}{2}$ | 6 | × | $\frac{3}{8}$ | ,, | 4 |
| $\frac{7}{8}$ | × | $\frac{1}{8}$ | ,, | $\frac{3}{4}$ | $2\frac{7}{8}$ | × | $\frac{1}{4}$ | ,, | $2\frac{1}{2}$ | $6\frac{1}{4}$ | × | $\frac{3}{8}$ | ,, | 4 |
| 1 | × | $\frac{1}{4}$ | ,, | $\frac{3}{4}$ | 3 | × | $\frac{1}{4}$ | ,, | $2\frac{3}{4}$ | $6\frac{1}{2}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $1\frac{1}{8}$ | × | $\frac{1}{4}$ | ,, | 1 | $3\frac{1}{4}$ | × | $\frac{1}{4}$ | ,, | $2\frac{3}{4}$ | $6\frac{3}{4}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $1\frac{1}{4}$ | × | $\frac{1}{4}$ | ,, | 1 | $3\frac{1}{2}$ | × | $\frac{1}{4}$ | ,, | 3 | 7 | × | $\frac{3}{8}$ | ,, | 4 |
| $1\frac{3}{8}$ | × | $\frac{1}{4}$ | ,, | 1 | $3\frac{3}{4}$ | × | $\frac{1}{4}$ | ,, | 3 | $7\frac{1}{4}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $1\frac{1}{2}$ | × | $\frac{1}{4}$ | ,, | 1 | 4 | × | $\frac{1}{4}$ | ,, | 3 | $7\frac{1}{2}$ | × | $\frac{3}{8}$ | ,, | 4 |
| $1\frac{5}{8}$ | × | $\frac{1}{4}$ | ,, | $1\frac{1}{2}$ | $4\frac{1}{4}$ | × | $\frac{3}{8}$ | ,, | 3 | $7\frac{3}{4}$ | × | $2\frac{1}{4}$ | ,, | 4 |
| $1\frac{3}{4}$ | × | $\frac{1}{4}$ | ,, | $1\frac{1}{2}$ | $4\frac{1}{2}$ | × | $\frac{3}{8}$ | ,, | 3 | 8 | × | $2\frac{1}{4}$ | ,, | 4 |
| $1\frac{7}{8}$ | × | $\frac{1}{4}$ | ,, | $1\frac{3}{4}$ | $4\frac{3}{4}$ | × | $\frac{3}{8}$ | ,, | 3 | | | | | |

### ROUND EDGE FLATS.

From 1 to 2 inches wide × $\frac{1}{8}$ to $\frac{5}{8}$ inch thick.

### SQUARES.

| ins. | ins. | ins. |
|---|---|---|
| $\frac{1}{4}$ | $\frac{5}{16}$ | $\frac{3}{8}$ |
| $\frac{7}{16}$ | $\frac{1}{2}$ | $\frac{9}{16}$ |
| $\frac{5}{8}$ | $\frac{11}{16}$ | $\frac{3}{4}$ |
| $\frac{13}{16}$ | $\frac{7}{8}$ | $\frac{15}{16}$ |
| 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ |
| $1\frac{3}{8}$ | $1\frac{1}{2}$ | $1\frac{5}{8}$ |
| $1\frac{3}{4}$ | $1\frac{7}{8}$ | 2 |
| $2\frac{1}{8}$ | $2\frac{1}{4}$ | $2\frac{3}{8}$ |
| $2\frac{1}{2}$ | $2\frac{5}{8}$ | $2\frac{3}{4}$ |
| $2\frac{7}{8}$ | 3 | $3\frac{1}{8}$ |
| $3\frac{1}{4}$ | $3\frac{1}{2}$ | $3\frac{3}{4}$ |
| 4 | $4\frac{1}{4}$ | $4\frac{1}{2}$ |
| $4\frac{3}{4}$ | 5 | $5\frac{1}{4}$ |
| $5\frac{1}{2}$ | $5\frac{3}{4}$ | 6 |

### ROUNDS.

| ins. | ins. | ins. | ins. |
|---|---|---|---|
| $\frac{1}{4}$ | $\frac{9}{32}$ | $\frac{5}{16}$ | $\frac{11}{32}$ |
| $\frac{3}{8}$ | $\frac{13}{32}$ | $\frac{7}{16}$ | $\frac{15}{32}$ |
| $\frac{31}{64}$ | $\frac{1}{2}$ | $\frac{17}{32}$ | $\frac{9}{16}$ |
| $\frac{19}{32}$ | $\frac{5}{8}$ | $\frac{21}{32}$ | $\frac{11}{16}$ |
| $\frac{23}{32}$ | $\frac{3}{4}$ | $\frac{25}{32}$ | $\frac{13}{16}$ |
| $\frac{27}{32}$ | $\frac{7}{8}$ | $\frac{29}{32}$ | $\frac{15}{16}$ |
| 1 | $1\frac{1}{32}$ | $1\frac{1}{16}$ | $1\frac{3}{32}$ |
| $1\frac{1}{8}$ | $1\frac{3}{16}$ | $1\frac{1}{4}$ | $1\frac{5}{16}$ |
| $1\frac{3}{8}$ | $1\frac{7}{16}$ | $1\frac{1}{2}$ | $1\frac{5}{8}$ |
| $1\frac{3}{4}$ | $1\frac{7}{8}$ | 2 | $2\frac{1}{8}$ |
| $2\frac{1}{4}$ | $2\frac{3}{8}$ | $2\frac{1}{2}$ | $2\frac{5}{8}$ |
| $2\frac{3}{4}$ | $2\frac{7}{8}$ | 3 | $3\frac{1}{8}$ |
| $3\frac{1}{4}$ | $3\frac{3}{8}$ | $3\frac{1}{2}$ | $3\frac{5}{8}$ |
| $3\frac{3}{4}$ | 4 | $4\frac{1}{4}$ | $4\frac{1}{2}$ |
| $4\frac{3}{4}$ | 5 | $5\frac{1}{4}$ | $5\frac{1}{2}$ |
| $5\frac{3}{4}$ | 6 | $6\frac{1}{4}$ | $6\frac{1}{2}$ |
| $6\frac{3}{4}$ | 7 | $7\frac{1}{2}$ | 8 |
| $8\frac{1}{2}$ | 9 | | |

### SLABS.

15 ins. to 20 ins. wide × 5 ins. to 8 ins. thick.
20 ,, ,, 36 ,, ,, × 5 ,, ,, 12 ,, ,,
36 ,, ,, 56 ,, ,, × 9 ,, ,, 12 ,, ,,

## MAKER'S EXTRAS ON SIEMENS' STEEL BOILER PLATES AND BARS.

### Marine, Land, and Locomotive Boiler Plates.

For every 3 inches or part over free limits in undernoted Table, 2s. 6d. per ton.

For every 3 inches or part over Width indicated in this Table, 2s. 6d. per ton.

| Thickness of plate . . | ins. $\frac{3}{8}$ and under $\frac{7}{16}$ | ins. $\frac{7}{16}$ and under $\frac{1}{2}$ | ins. $\frac{1}{2}$ and under $\frac{9}{16}$ | ins. $\frac{9}{16}$ and under $\frac{5}{8}$ | ins. $\frac{5}{8}$ and under $\frac{3}{4}$ | ins. $\frac{3}{4}$ and under $\frac{7}{8}$ | ins. $\frac{7}{8}$ and under 1 | ins. 1 and under $1\frac{1}{4}$ | ins. $1\frac{1}{4}$ and under $1\frac{1}{2}$ | $1\frac{1}{2}$ |
|---|---|---|---|---|---|---|---|---|---|---|
| Width supplied free of extra for width . . | 75 | 84 | 90 | 96 | 96 | 96 | 96 | 96 | 96 | 96 |

Weight over 80 cwts., 5s. per 5 cwts. or part up to 120 cwts., thereafter 10s. for every 5 cwts. or part thereof. For example, the extra for plates of 6 tons 3 cwts. would be £2, 10s.

All plates which are not rectangular will be considered sketches. Of these 15 per cent. will be allowed free, any in excess of 15 per cent. being charged . . . . . . 25s. per ton.
Over 1½ inches to 1⅝ inches thick inclusive . . 10s. ,,
,, 1⅝ inches to 1¾ inches ,, . . . 20s. ,,
,, 1¾ inches to 2 inches ,, . . . 30s. ,,

For the undermentioned minimum tensile strains:—
A minimum tensile strain of not more than 28½ tons, 5s. ,,
,, ,, ,, 29 ,, 10s. ,,
,, ,, ,, 30 ,, 20s. ,,
,, ,, ,, 31 ,, 40s. ,,
,, ,, ,, 32 ,, 60s. ,,
,, ,, ,, 33 ,, 80s. ,,
,, ,, ,, 34 ,, 100s. ,,
,, ,, ,, 35 ,, 120s. ,,

For the undermentioned low tensile strains—
Plates specified not to exceed 27 tons per square inch, 10s. ,,
,, ,, 26 ,, ,, 20s. ,,
,, ,, 25 ,, ,, 40s. ,,

| | | |
|---|---|---|
| A minimum elongation, 25 per cent. on 10-inch . | 20s. | per ton. |
| ,, ,, 25 ,, 8-inch . | 15s. | ,, |
| Plates 20 feet long or over, and at the same time 6 feet wide or over, or plates 50 cwts. each or over, to pass Board of Trade Survey, or equal thereto . | 10s. | ,, |
| Admiralty Survey, or equal thereto . . . . . | 20s. | ,, |
| When the phosphorus and sulphur limit is ·035 . | 70s. | ,, |
| In every case where tensile limit narrowed to 2 tons . | 20s. | ,, |
| In every case where the range of tensile is over 2 and not exceeding 3 tons . . . . . . . | 10s. | ,, |

Testing fees, over and above Lloyd's fees, or where they exceed 1s. per ton, will be charged extra in full.

## Extras on Bars.

### ROUNDS AND SQUARES.

Rivet steel bars, $\frac{39}{64}$ inch to be taken at basis price.

| | | |
|---|---|---|
| Under $\frac{5}{8}$ inch to $\frac{1}{2}$ inch . . . . . . . | 5s. | per ton. |
| ,, $\frac{1}{2}$ inch to $\frac{7}{16}$ inch . . . . . | 10s. | ,, |
| ,, $\frac{7}{16}$ inch to $\frac{3}{8}$ inch . . . . . | 20s. | ,, |
| Over 3 inches to $3\frac{3}{4}$ inches . . . . . | 5s. | ,, |

$3\frac{7}{8}$ inches and up, subject to arrangement.

| | | |
|---|---|---|
| Stay bars to pass Board of Trade or Admiralty Survey | 20s. | ,, |
| ,, ,, Lloyd's or Bureau Veritas Survey | 10s. | ,, |
| Turning Quality . . . . . . . . | 20s. | ,, |
| Double Survey . . . . . . . . | 1s. | ,, extra |

Ordinary cut lengths 5 feet and over 3 feet, 5s. per ton extra. 3 feet to 2 feet, 10s. per ton extra. Special extras will be charged for cutting to short lengths, or to exact length, and for special straightening.

Extras for testing fees and for restricted tensile limits, the same for bars as for plates.

## Plain Rolled Steel Shafts.

(As supplied by J. Spencer & Son, Newburn.)

| | | | | |
|---|---|---|---|---|
| 10 inches | to | $10\frac{1}{2}$ inches | 30 to 35 | feet long. |
| 11 ,, | | $11\frac{1}{2}$ ,, | 26 ,, 30 | ,, |
| 12 ,, | | $12\frac{1}{2}$ ,, | 22 ,, 25 | ,, |
| 13 ,, | | $13\frac{1}{2}$ ,, | 35 ,, 40 | ,, |
| 14 ,, | | $14\frac{1}{2}$ ,, | 30 | ,, |
| 15 ,, | | $15\frac{1}{2}$ ,, | 25 ,, 30 | ,, |
| 16 ,, | | $16\frac{1}{2}$ ,, | 20 ,, 25 | ,, |

These shafts are now supplied with flanges formed by upsetting, etc., by hydraulic presses.

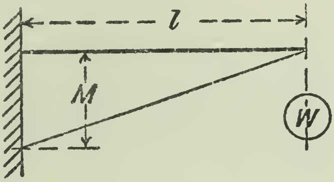
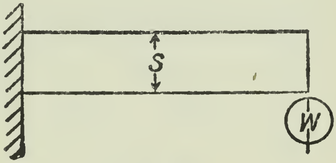
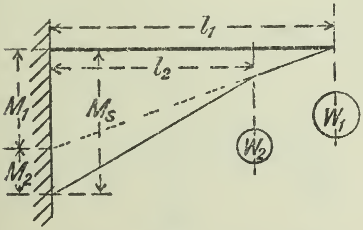
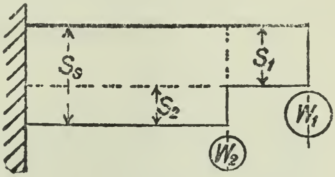
## Table CXXXV.—Limits of Safe Working Stresses on Various Metals (Tensile Continuous Service).

| Metal or Alloy. | Nature of the Load. | At revolutions per minute, not exceeding | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 60 | 100 | 200 | 300 | 400 | 500 | 600 | 700 | 800 | 1000 |
| Iron castings of good quality, | Steady | 5,150 | 5,000 | 4,800 | 4,700 | 4,600 | 4,550 | 4,500 | 4,460 | 4,450 | 4,440 |
| | Intermittent | 4,700 | 4,500 | 4,200 | 3,940 | 3,750 | 3,600 | 3,440 | 3,300 | 3,200 | 3,000 |
| | Alternating | 3,900 | 3,700 | 3,350 | 3,100 | 2,900 | 2,700 | 2,550 | 2,400 | 2,300 | 2,200 |
| Mild steel castings, good quality, | Steady | 15,200 | 14,650 | 13,800 | 13,100 | 12,550 | 12,050 | 11,600 | 11,200 | 10,800 | 10,150 |
| | Intermittent | 13,450 | 12,800 | 11,950 | 11,250 | 10,650 | 10,150 | 9,650 | 9,200 | 8,850 | 8,200 |
| | Alternating | 11,150 | 10,550 | 9,750 | 8,950 | 8,400 | 7,800 | 7,330 | 6,900 | 6,500 | 6,000 |
| Mild steel bars and plates, | Steady | 16,800 | 16,500 | 15,900 | 15,450 | 15,100 | 14,800 | 14,550 | 14,350 | 14,150 | 13,800 |
| | Intermittent | 14,350 | 14,000 | 13,400 | 12,900 | 12,500 | 12,200 | 11,930 | 11,700 | 11,500 | 11,150 |
| | Alternating | 12,100 | 11,750 | 11,100 | 10,600 | 10,250 | 9,900 | 9,600 | 9,360 | 9,150 | 8,800 |
| High tensile steel bars and forgings, nickel, chrome, vanadium, | Steady | 26,800 | 26,350 | 25,500 | 24,800 | 24,260 | 23,800 | 23,350 | 22,950 | 22,550 | 21,750 |
| | Intermittent | 24,350 | 23,850 | 22,950 | 22,250 | 21,700 | 21,200 | 20,750 | 20,300 | 19,900 | 19,100 |
| | Alternating | 22,100 | 21,650 | 20,700 | 20,000 | 19,420 | 18,900 | 18,450 | 18,050 | 17,700 | 16,950 |
| Bronze, or gun-metal, Admiralty alloy, | Steady | 7,800 | 7,600 | 7,200 | 6,930 | 6,700 | 6,500 | 6,350 | 6,200 | 6,100 | 5,950 |
| | Intermittent | 7,000 | 6,800 | 6,350 | 6,000 | 5,660 | 5,400 | 5,150 | 4,930 | 4,720 | 4,350 |
| | Alternating | 5,900 | 5,600 | 5,100 | 4,650 | 4,300 | 4,000 | 3,750 | 3,500 | 3,300 | 3,000 |
| High tensile bronzes, carefully cast, | Steady | 12,700 | 12,500 | 12,100 | 11,700 | 11,550 | 11,250 | 11,050 | 10,850 | 10,700 | 10,400 |
| | Intermittent | 11,450 | 11,000 | 10,300 | 9,800 | 9,350 | 9,000 | 8,700 | 8,400 | 8,150 | 7,500 |
| | Alternating | 9,500 | 9,000 | 8,300 | 7,750 | 7,300 | 6,930 | 6,600 | 6,300 | 6,000 | 5,450 |
| High tensile bronze bars, &c., unannealed, | Steady | 22,500 | 21,600 | 20,800 | 20,100 | 19,600 | 19,200 | 18,900 | 18,700 | 18,500 | 18,140 |
| | Intermittent | 19,500 | 19,000 | 18,000 | 17,200 | 16,600 | 16,030 | 15,500 | 15,000 | 14,500 | 13,600 |
| | Alternating | 15,500 | 15,000 | 14,000 | 13,200 | 12,560 | 12,000 | 11,500 | 11,000 | 11,600 | 10,750 |
| High tensile bronze bars, &c., annealed, | Steady | 15,250 | 15,000 | 14,520 | 14,040 | 13,860 | 13,500 | 13,250 | 13,020 | 12,860 | 12,500 |
| | Intermittent | 13,750 | 13,200 | 12,360 | 11,760 | 11,120 | 10,800 | 10,440 | 10,089 | 9,800 | 9,100 |
| | Alternating | 11,400 | 10,900 | 9,960 | 9,300 | 8,760 | 8,320 | 7,920 | 7,560 | 7,250 | 6,750 |

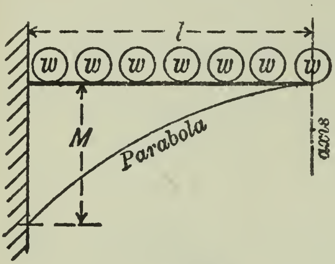
For shearing stresses, multiply the above by 1·62 for cast iron; 0·8 for steel; strong zinc bronzes, 0·60; delta metal, 0·7; phosphor bronze, cast, 1·10; rolled, 0·63.

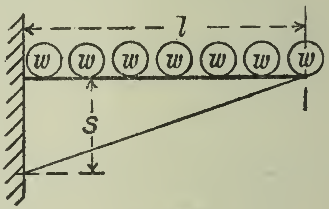
## BENDING MOMENTS, Etc.

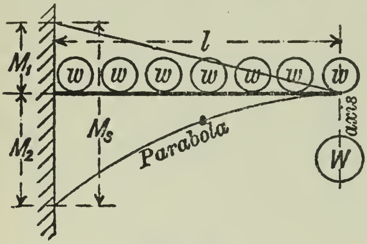
### Table CXXXVI.—Graphic Representation of Bending Moments and Shearing Forces in Simple Cases.

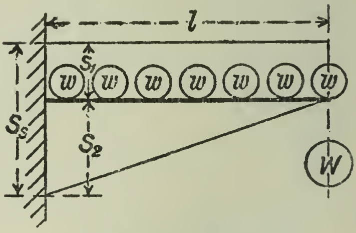
<table>
<tr><th>Bending Moment.</th><th>Shearing Force.</th></tr>
<tr><td colspan="2">$f$ = greatest permissible stress on material; $z$ = modulus of section —<i>see</i> Table CI.; M = greatest bending moment; S = greatest shearing force.</td></tr>
<tr><td colspan="2"><b>Beams fixed at one end.</b></td></tr>
<tr><td> <br>$M = Wl$<br>Working load $= \frac{fz}{l}$</td><td> <br>$S = W$</td></tr>
<tr><td> <br>$M_1 = W_1 l_1$; and $M_2 = W_2 l_2$<br>$M_s = W_1 l_1 + W_2 l_2$</td><td> <br>$S_1 = W_1$; and $S_2 = W_2$<br>$S_s = W_1 + W_2$</td></tr>
</table>

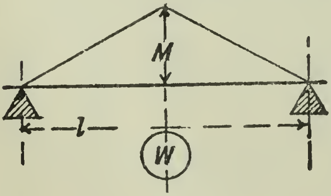
## Beams fixed at one end—*continued.*

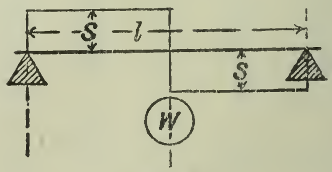
<table>
<tr><th>Bending Moment.</th><th>Shearing Force.</th></tr>
<tr><td>

$M = \frac{wl^2}{2}$<br>Working load (per unit of length) $= \frac{2fz}{l^2}$</td><td>

$S = wl$</td></tr>
<tr><td>

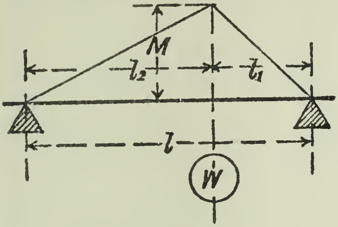
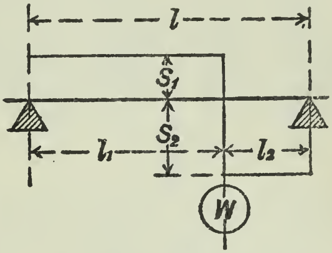
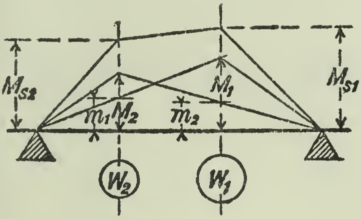
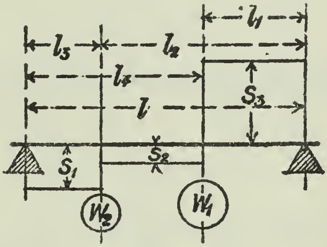
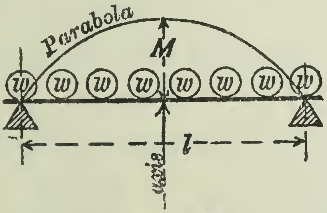
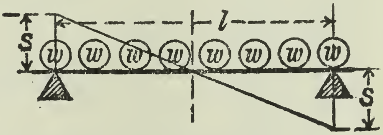
$M_1 = Wl$<br>$M_2 = \frac{wl}{2}$<br>$M_s = Wl + \frac{wl}{2}$</td><td>

$S_1 = W$ ; and $S_2 = wl$<br>$S_s = W + wl$</td></tr>
<tr><th colspan="2">Beams supported at both ends.</th></tr>
<tr><td>

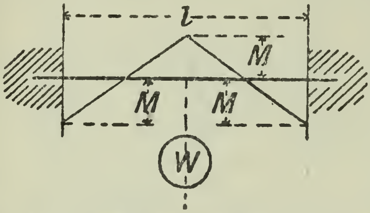
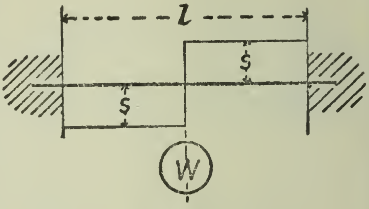
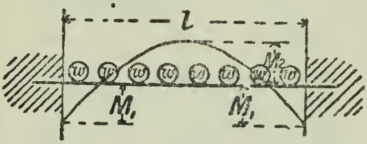
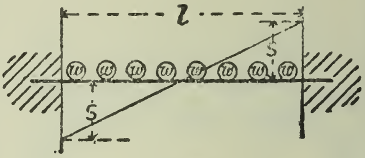
$M = W\frac{l}{4}$<br>Working load $= 4\frac{fz}{l}$</td><td>

$S = \frac{W}{2}$</td></tr>
</table>

## Beams supported at both ends—*continued.*

| **Bending Moment.** | **Shearing Force.** |
|---|---|
|  $M = W\frac{l_1 l_2}{l}$<br>Working load $=\frac{fzl}{l_1 l_2}$ |  $S_1 = W\frac{l_2}{l}$; $S_2 = W\frac{l_1}{l}$ |
|  $M_{s1} = M_1 + m_2$<br>$M_{s2} = M_2 + m_1$ |  $S_1 = \frac{W_1 l_1 + W_2 l_2}{l}$<br>$S_2 = \frac{W_1 l_1 + W_2 l_2}{l} - W_2$<br>$S_3 = \frac{W_1 l_1 + W_2 l_2}{l} - (W_1 + W_2)$<br>$= -\frac{W_1 l_4 + W_2 l_3}{l}$ |
|  $M = \frac{wl^2}{8}$<br>Working load (per unit of length) $=\frac{8fz}{l^2}$ |  $S = \frac{wl}{2}$ |

## Beams fixed at both ends.

| Bending Moment. | Shearing Force. |
|---|---|
|  <br>In this case the bending moments at M, M and M are equal, and have the value $M=\frac{Wl}{8}$. Working load $=\frac{8fz}{l}$. |  <br>$S=\frac{W}{2}$. |
|  <br>Bending moments at ends (M₁)<br>$M_1=\frac{wl^2}{12}$.<br>Bending moment at centre (M₂)<br>$M_2=\frac{wl^2}{24}$.<br>Working load (per unit of length) $=\frac{12fz}{l^2}$. |  <br>$S=\frac{wl}{2}$. |

## Deflection of Beams of Uniform Section.

$D=\frac{1}{3}\left(\frac{WL^3}{EI}\right)$ for a beam fixed at one end and loaded at the other.

$D=\frac{1}{8}\left(\frac{WL^3}{EI}\right)$ „ „ „ „ uniformly.

$D=\frac{1}{192}\left(\frac{WL^3}{EI}\right)$ „ fixed at both ends and loaded in the middle.

$D=\frac{1}{384}\left(\frac{WL^3}{EI}\right)$ „ „ „ „ uniformly.

$D=\frac{1}{48}\left(\frac{WL^3}{EI}\right)$ „ supported at both ends and loaded in the middle.

$D=\frac{5}{384}\left(\frac{WL^3}{EI}\right)$ „ „ „ „ uniformly.

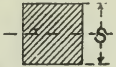
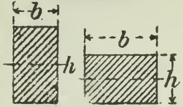
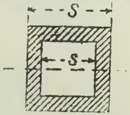
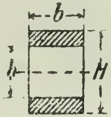
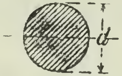
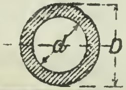
Where D is deflection in inches; W the total load in lbs.; L the length in inches; E the modulus of elasticity; and I the moment of inertia of section.

Average value of E is as follows:—For cast-iron, 17,000,000; wrought-iron, 27,000,000; wrought-steel, 29,000,000.

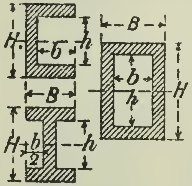
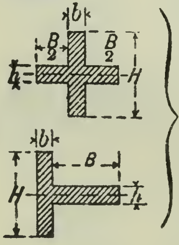
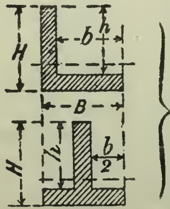
For values of I see page 467, 468.

### Table CXXXVII.—Moment of Inertia, Modulus, &c., of some Sections.

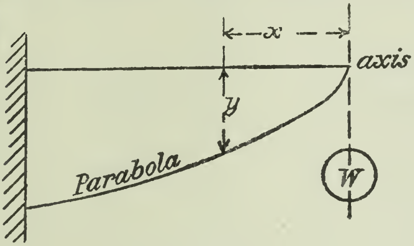
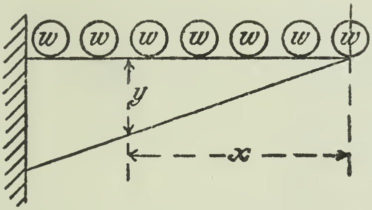
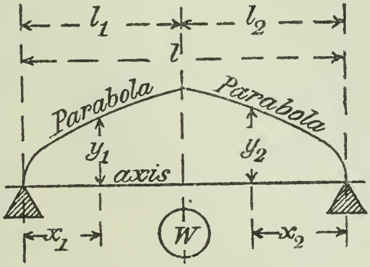
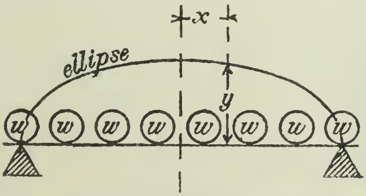
The plane of bending is supposed perpendicular to plane of paper, and parallel to side of page.

| Form of section. | Area of section. A | Moment of inertia of section about axis through centre of gravity. I | Square of rads. of gyration of section. $\rho^2 = \frac{I}{A}$ | "Modulus" of section. $Z = \frac{I}{y}$ |
|---|---|---|---|---|
|  | $s^2$ | $\frac{s^4}{12}$ | $\frac{s^2}{12}$ | $\frac{s^3}{6}$ |
|  | $bh$ | $\frac{bh^3}{12}$ | $\frac{h^2}{12}$ | $\frac{bh^2}{6}$ |
|  | $S^2 - s^2$ | $\frac{S^4 - s^4}{12}$ | $\frac{S^2 + s^2}{12}$ | $\frac{1}{6}\left(\frac{S^4 - s^4}{S}\right)$ |
|  | $b\,(H - h)$ | $\frac{b\,(H^3 - h^3)}{12}$ | $\frac{H^2 + hH + h^2}{12}$ | $\frac{b\,(H^3 - h^3)}{6H}$ |
|  | $\cdot 7854 d^2$ | $\cdot 0491 d^4$ | $\frac{d^2}{16}$ | $\cdot 0982 d^3$ |
|  | $\cdot 7854\,(D^2 - d^2)$ | $\cdot 0491\,(D^4 - d^4)$ | $\frac{D^2 + d^2}{16}$ | $\cdot 0982\left(\frac{D^4 - d^4}{D}\right)$ |

## Table CXXXVII.—Moment of Inertia, Modulus, &c., of some Sections—*continued*.

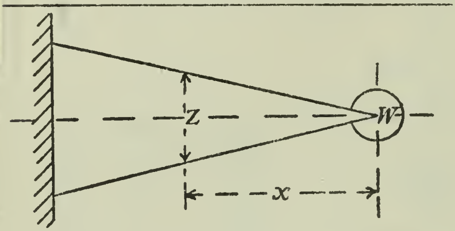
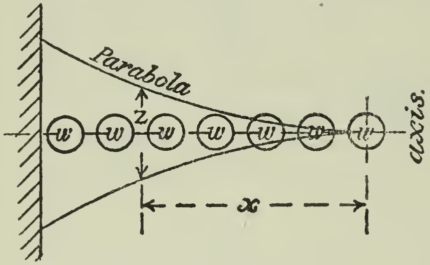
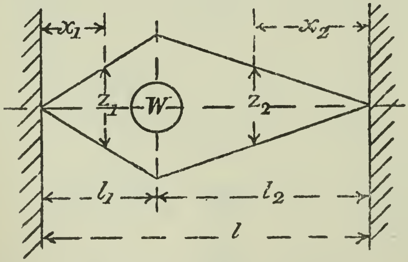
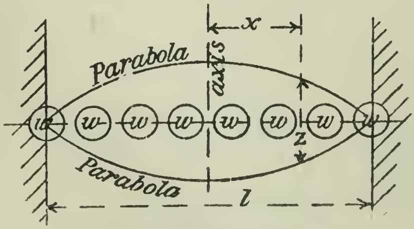
| Form of section. | Area of section. A | Moment of inertia of section about axis through centre of gravity. I | Square of rads. of gyration of section. $\rho^2=\frac{I}{A}$ | "Modulus" of section. $Z=\frac{I}{y}$ |
|---|---|---|---|---|
|  H<br>b<br>h<br>B<br>H<br>b<br>h<br>H<br>$\frac{b}{2}$<br>B<br>h | $BH - bh$ | $\frac{BH^3 - bh^3}{12}$ | $\frac{1}{12}\left(\frac{BH^3 - bh^3}{BH - bh}\right)$ | $\frac{BH^3 - bh^3}{6H}$ |
|  b<br>$\frac{B}{2}$<br>$\frac{B}{2}$<br>h<br>H<br>b<br>B<br>H<br>h | $Bh + bH$ | $\frac{Bh^3 + bH^3}{12}$ | — | $\frac{Bh^3 + bH^3}{6H}$ |
|  | $BH - bh$ | $\frac{(BH^2 - bh^2)^2 - 4BHbh(H - h)^2}{12(BH - bh)}$ | — | $\frac{(BH^2 - bh^2)^2 - 4BHbh(H - h)^2}{6(BH^2 - bh^2)}$ |

## Table CXXXVIII.—Forms of Beams of Uniform Strength.

| Breadth ($b$) uniform throughout. | |
|---|---|
| **Elevation of arrangement.** | **Equation for dimensions.** |
|  | $y = \sqrt{\frac{6Wx}{bf}}$ |
|  | $y = \sqrt{\frac{3wx^2}{bf}}$ |
|  l1<br>l2<br>l<br>Parabola<br>Parabola<br>y1<br>axis<br>y2<br>x1<br>W<br>x2 | $y_1 = \sqrt{\frac{6Wl_2x_1}{bfl}}$<br>$y_2 = \sqrt{\frac{6Wl_1x_2}{bfl}}$ |
|  | $y = \sqrt{\frac{3w(l^2 - 4x^2)}{4bf}}$ |

## Table CXXXVIII.—Forms of Beams of Uniform Strength—*continued.*

Depth ($h$) uniform throughout.

| Plan of arrangement. | Equation for dimensions. |
|---|---|
|  | $z=\frac{6Wx}{h^2f}$ |
|  Parabola<br>z<br>w<br>w<br>w<br>w<br>w<br>w<br>w<br>axis.<br>x | $z=\frac{3wx^2}{h^2f}$ |
|  $x_1$<br>$x_2$<br>$z_1$<br>W<br>$z_2$<br>$l_1$<br>$l_2$<br>$l$ | $z_1=\frac{6Wl_2x_1}{h^2fl}$<br>$z_2=\frac{6Wl_1x_2}{h^2fl}$ |
|  x<br>Parabola<br>axis<br>w<br>w<br>w<br>w<br>w<br>w<br>w<br>w<br>z<br>Parabola<br>l | $z=\frac{3w(l^2-4x^2)}{4h^2f}$ |

## Table CXXXIX.—The Greatest Fluid Test Pressure that should be put on Flat Surfaces of Good Cast Iron in lbs. per sq. inch.

| Thickness, Inch. | Breadth of Strip in Inches. | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| 0·3 | 304 | 135 | 76 | 48 | 34 | 25 | 19 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·4 | 540 | 240 | 135 | 86 | 60 | 44 | 34 | 27 | 22 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·5 | 944 | 375 | 238 | 135 | 93 | 69 | 53 | 42 | 34 | 28 | 23 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·6 | 1215 | 540 | 304 | 194 | 135 | 99 | 76 | 60 | 49 | 40 | 34 | 29 | 25 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·7 | 1654 | 735 | 413 | 265 | 184 | 135 | 103 | 82 | 66 | 55 | 49 | 40 | 34 | 30 | 26 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·8 | 2160 | 960 | 540 | 346 | 240 | 176 | 135 | 107 | 86 | 71 | 60 | 51 | 44 | 38 | 34 | 30 | 26 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·9 | 2734 | 1215 | 683 | 437 | 304 | 223 | 171 | 135 | 109 | 90 | 76 | 64 | 56 | 48 | 42 | 37 | 34 | 31 | 27 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 1·0 | 3375 | 1500 | 844 | 540 | 372 | 276 | 211 | 166 | 135 | 112 | 94 | 80 | 69 | 60 | 52 | 47 | 42 | 38 | 34 | 30 | 27 | ... | ... | ... | ... | ... | ... | ... | ... |
| 1·1 | 4082 | 1815 | 1020 | 653 | 454 | 333 | 255 | 202 | 163 | 135 | 114 | 99 | 83 | 72 | 64 | 56 | 50 | 45 | 41 | 37 | 34 | 31 | 28 | ... | ... | ... | ... | ... | ... |
| 1·2 | ... | 2160 | 1215 | 778 | 540 | 397 | 304 | 240 | 194 | 161 | 135 | 115 | 99 | 86 | 76 | 67 | 60 | 54 | 49 | 44 | 40 | 37 | 34 | 31 | ... | ... | ... | ... | ... |
| 1·3 | ... | 2534 | 1425 | 912 | 633 | 465 | 356 | 282 | 228 | 188 | 158 | 135 | 116 | 101 | 89 | 79 | 70 | 63 | 57 | 51 | 47 | 43 | 40 | 37 | 34 | ... | ... | ... | ... |
| 1·4 | ... | 2940 | 1654 | 1058 | 702 | 540 | 414 | 327 | 265 | 219 | 184 | 157 | 135 | 118 | 103 | 91 | 81 | 73 | 66 | 60 | 55 | 50 | 46 | 42 | 39 | 36 | 34 | ... | ... |
| 1·5 | ... | 3374 | 1899 | 1215 | 833 | 620 | 475 | 374 | 304 | 251 | 211 | 180 | 155 | 135 | 119 | 105 | 94 | 84 | 76 | 69 | 63 | 57 | 52 | 48 | 43 | 40 | 38 | 36 | 34 |
| 1·6 | ... | 3840 | 2160 | 1382 | 960 | 705 | 540 | 427 | 346 | 286 | 240 | 204 | 177 | 153 | 135 | 120 | 107 | 95 | 86 | 78 | 71 | 65 | 60 | 55 | 51 | 47 | 44 | 41 | 38 |
| 1·7 | ... | 4335 | 2438 | 1561 | 1084 | 796 | 609 | 482 | 390 | 313 | 271 | 231 | 200 | 174 | 153 | 135 | 120 | 108 | 97 | 89 | 81 | 74 | 68 | 62 | 58 | 54 | 50 | 46 | 43 |
| 1·8 | ... | ... | 2734 | 1750 | 1215 | 893 | 683 | 540 | 437 | 361 | 304 | 260 | 223 | 195 | 171 | 151 | 135 | 121 | 109 | 99 | 90 | 83 | 76 | 70 | 65 | 60 | 56 | 52 | 49 |
| 1·9 | ... | ... | 3046 | 1949 | 1354 | 996 | 761 | 602 | 487 | 403 | 330 | 288 | 250 | 216 | 190 | 169 | 150 | 135 | 122 | 110 | 100 | 92 | 84 | 78 | 72 | 67 | 62 | 58 | 54 |
| 2·0 | ... | ... | 3374 | 2160 | 1500 | 1102 | 844 | 667 | 540 | 446 | 357 | 320 | 276 | 240 | 211 | 187 | 166 | 150 | 135 | 122 | 112 | 102 | 94 | 86 | 80 | 74 | 69 | 64 | 60 |

If of high-class metal (9 tons tensile), multiply by 1·4.

For a Steady Load the maximum Working Pressure should not exceed 60 per cent. of the above; for an Intermittent Load, 46 per cent.; and for an Alternating, 35 per cent. If made of the best iron and twice melted, the pressures may be 33 per cent. higher.

## Table CXL.—The Greatest Fluid Test Pressure on Flat Surfaces of Rolled Mild Steel or Strong Bronzes when unsupported.

| Thickness in 32nds of an inch. | Breadth of Strips in Inches. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 4. | 5. | 6. | 7. | 8. | 9. | 10. | 11. | 12. | 13. | 14. | 15. | 16. | 17. | 18. | 19. |
| 3 | 28 | 18 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 4 | 50 | 32 | 22 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 5 | 78 | 50 | 35 | 25 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 6 | 112 | 72 | 50 | 37 | 28 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 7 | 153 | 98 | 68 | 50 | 38 | 30 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 8 | 200 | 128 | 89 | 65 | 50 | 40 | 32 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 9 | 253 | 162 | 112 | 83 | 63 | 50 | 40 | 34 | ... | ... | ... | ... | ... | ... | ... | ... |
| 10 | 312 | 200 | 139 | 102 | 78 | 62 | 50 | 41 | 35 | ... | ... | ... | ... | ... | ... | ... |
| 11 | 378 | 242 | 168 | 123 | 94 | 75 | 60 | 50 | 42 | 36 | ... | ... | ... | ... | ... | ... |
| 12 | 450 | 288 | 200 | 147 | 112 | 89 | 72 | 60 | 50 | 42 | 37 | ... | ... | ... | ... | ... |
| 13 | 528 | 338 | 235 | 172 | 132 | 104 | 85 | 70 | 59 | 50 | 43 | 38 | ... | ... | ... | ... |
| 14 | 612 | 392 | 271 | 200 | 153 | 121 | 98 | 81 | 68 | 58 | 50 | 43 | 38 | ... | ... | ... |
| 15 | ... | 450 | 312 | 238 | 176 | 139 | 113 | 93 | 78 | 67 | 57 | 50 | 44 | 39 | ... | ... |
| 16 | ... | 512 | 355 | 261 | 200 | 158 | 128 | 106 | 89 | 76 | 65 | 57 | 50 | 44 | 40 | ... |
| 17 | ... | 578 | 401 | 295 | 226 | 179 | 145 | 119 | 100 | 85 | 74 | 65 | 56 | 50 | 45 | 40 |
| 18 | ... | ... | 450 | 330 | 253 | 200 | 162 | 134 | 112 | 96 | 83 | 72 | 63 | 56 | 50 | 45 |
| 19 | ... | ... | 501 | 368 | 282 | 223 | 181 | 149 | 125 | 107 | 92 | 80 | 70 | 63 | 56 | 50 |
| 20 | ... | ... | 555 | 409 | 312 | 247 | 200 | 166 | 139 | 118 | 102 | 89 | 78 | 70 | 62 | 55 |
| 21 | ... | ... | ... | 450 | 344 | 272 | 221 | 182 | 153 | 130 | 112 | 98 | 86 | 76 | 68 | 61 |
| 22 | ... | ... | ... | 493 | 378 | 298 | 242 | 200 | 168 | 143 | 123 | 107 | 94 | 83 | 74 | 67 |
| 23 | ... | ... | ... | 540 | 413 | 327 | 265 | 219 | 184 | 156 | 135 | 117 | 103 | 91 | 81 | 73 |
| 24 | ... | ... | ... | ... | 450 | 356 | 288 | 238 | 200 | 170 | 147 | 128 | 112 | 100 | 89 | 79 |

*N.B.*—When secured at the edges or stiffened by flanges or continuous, multiply by 2. The Working Pressure with a Steady Load may be 60 per cent. of the test; for an Intermittent Load, 46 per cent.; and for Alternating, 35 per cent.

Table CXLI.—The Greatest Fluid Test Pressure that should be put on Flat Surfaces of Phosphor Bronze Castings in lbs. per sq. inch.

| Thickness, Inch. | Breadth of Strip in Inches. | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| 0·15 | 140 | 62 | 35 | 22 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·20 | 250 | 111 | 62 | 40 | 28 | 20 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·25 | 390 | 174 | 97 | 62 | 48 | 32 | 24 | 20 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·30 | 563 | 250 | 141 | 90 | 63 | 46 | 35 | 28 | 23 | 19 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·35 | 781 | 347 | 195 | 125 | 87 | 63 | 49 | 39 | 31 | 24 | 20 | 18 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·40 | 1000 | 444 | 250 | 160 | 111 | 82 | 62 | 49 | 40 | 33 | 28 | 24 | 20 | 18 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·45 | 1262 | 561 | 315 | 202 | 140 | 103 | 79 | 62 | 50 | 42 | 34 | 30 | 26 | 23 | 20 | 18 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·50 | 1562 | 694 | 390 | 250 | 173 | 128 | 98 | 77 | 63 | 52 | 43 | 37 | 32 | 28 | 24 | 21 | 19 | 17 | 16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·55 | 1890 | 840 | 473 | 302 | 210 | 154 | 118 | 93 | 75 | 62 | 52 | 45 | 38 | 34 | 30 | 26 | 23 | 21 | 19 | 17 | 16 | ... | ... | ... | ... | ... | ... | ... | ... |
| 0·60 | 2250 | 1000 | 562 | 360 | 250 | 183 | 140 | 111 | 90 | 73 | 62 | 53 | 46 | 40 | 35 | 31 | 28 | 25 | 23 | 21 | 19 | 17 | 16 | ... | ... | ... | ... | ... | ... |
| 0·65 | 2640 | 1174 | 660 | 422 | 294 | 215 | 165 | 130 | 106 | 87 | 73 | 62 | 54 | 47 | 41 | 36 | 32 | 29 | 26 | 24 | 22 | 20 | 19 | 18 | 16 | ... | ... | ... | ... |
| 0·70 | 3056 | 1358 | 764 | 489 | 339 | 250 | 191 | 151 | 122 | 101 | 85 | 72 | 62 | 54 | 48 | 42 | 38 | 34 | 30 | 27 | 25 | 23 | 21 | 20 | 19 | 18 | 16 | ... | ... |
| 0·75 | 3512 | 1562 | 878 | 562 | 390 | 267 | 219 | 173 | 141 | 116 | 98 | 83 | 72 | 62 | 55 | 49 | 44 | 39 | 35 | 32 | 29 | 26 | 24 | 22 | 20 | 19 | 18 | 17 | 16 |
| 0·80 | 4000 | 1778 | 1000 | 640 | 444 | 327 | 250 | 198 | 160 | 132 | 111 | 95 | 82 | 71 | 62 | 55 | 49 | 44 | 40 | 36 | 33 | 30 | 28 | 26 | 24 | 22 | 20 | 19 | 18 |
| 0·85 | ... | 2007 | 1129 | 722 | 502 | 379 | 282 | 223 | 181 | 149 | 125 | 107 | 92 | 80 | 70 | 62 | 56 | 50 | 45 | 41 | 37 | 34 | 31 | 29 | 27 | 25 | 23 | 21 | 20 |
| 0·90 | ... | 2250 | 1266 | 810 | 563 | 418 | 316 | 250 | 203 | 167 | 141 | 120 | 103 | 90 | 79 | 70 | 62 | 56 | 51 | 46 | 42 | 38 | 35 | 32 | 30 | 28 | 26 | 24 | 22 |
| 0·95 | ... | 2507 | 1410 | 903 | 627 | 460 | 352 | 277 | 226 | 187 | 157 | 133 | 115 | 100 | 88 | 78 | 70 | 63 | 56 | 51 | 47 | 43 | 39 | 36 | 33 | 31 | 29 | 27 | 25 |
| 1·00 | ... | 2778 | 1562 | 1000 | 700 | 511 | 391 | 309 | 250 | 207 | 174 | 150 | 122 | 111 | 99 | 86 | 77 | 69 | 63 | 59 | 52 | 48 | 44 | 40 | 37 | 34 | 32 | 30 | 28 |
| 1·10 | ... | 3361 | 1891 | 1210 | 840 | 617 | 473 | 373 | 303 | 250 | 210 | 180 | 155 | 134 | 118 | 105 | 93 | 84 | 76 | 69 | 63 | 59 | 53 | 49 | 46 | 42 | 39 | 36 | 34 |
| 1·20 | ... | 4000 | 2250 | 1440 | 1000 | 735 | 563 | 444 | 360 | 298 | 250 | 213 | 184 | 160 | 140 | 125 | 111 | 100 | 90 | 82 | 75 | 68 | 63 | 58 | 54 | 50 | 46 | 43 | 40 |

For a Steady Load the maximum Working Pressure should not exceed 60 per cent. of the above; for an Intermittent Load, 46 per cent.; and for an Alternating, 33 per cent. Admiralty Bronze castings should not be tested beyond 75 per cent. of the above.

## EXPANSION OF METALS BY HEAT.

The following table shows approximately the extent to which metals, &c., expand under the action of heat:—

Table CXLII.—Expansion of Metals, &c., for rise of Temperature of 180° F.

| Material. | Change of length. | |
|---|---|---|
| | Fraction of total length. | Inches per foot. (C). |
| Cast-iron, . . . . | ·00117 | ·0140 |
| Wrought-iron, . . . | ·00122 | ·0146 |
| Steel, . . . . . | ·00120 | ·0146 |
| Copper, . . . . | ·00182 | ·0218 |
| Gun-metal, . . . . | ·00187 | ·0224 |
| Fire-brick, . . . . | ·000423 | ·0050 |

If $t_1$ and $t_2$ be the highest and lowest temperatures to which the object is exposed, the *alteration* in length is given by,—

$$\text{Alteration in length (increase or diminution)} = C \times \left(\frac{t_1 - t_2}{180}\right) \times L$$

where L is length in feet, value of C is taken from above table. and alteration of length is given in inches.

The expansion of metals, per degree rise of temperature, increases slightly as higher temperatures are reached, but for all practical purposes it may be assumed to be constant.

**Effect of temperature on strengths of metals.**—At 400° F. almost all alloys of copper, tin, and zinc lose from 15 to 20 per cent. of their ultimate strength ; but beyond this temperature some of the copper-zinc and copper-aluminium alloys weaken very rapidly.

Copper is similarly affected, and loses 8 to 10 per cent. of its strength at 400° F.

Cast-iron is practically unaffected up to 400° F.

Wrought-iron and mild steel *gain* from 10 to 15 per cent. in ultimate strength at 400° F.

Up to 400° F. the changes seem to progress regularly with the changes of temperature.

## Table CXLIII.—Effect of Temperature on Metals.

| Metal or Alloy. | At 100° F. | | At 250° F. | | At 350° F. | | At 400° F. | | At 450° F. | | Melting point. F. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Ultimate strength. | Extension. | Ultimate strength. | Extension. | Ultimate strength. | Extension. | Ultimate strength. | Extension. | Ultimate strength. | Extension. | |
| | | Per cent. | | Per cent. | | Per cent. | | Per cent. | | Per cent. | |
| Copper (0·2 per cent. arsenic), . | 30,325 | 30·3 | 31,000 | 26·2 | 30,950 | 21·1 | 29,750 | 27·0 | 29,000 | 29·0 | 2,000° |
| Admiralty bronze, cast, . . | 32,000 | 12·0 | 30,800 | 9·3 | 29,800 | 8·25 | 15,700 | 0·75 | 15,500 | 0·2 | 1,700° |
| Mercantile bronze (soft), cast, . | 29,000 | 26·0 | 28,800 | 25·8 | 28,000 | 23·0 | 27,750 | 25·0 | 9,700 | 1·2 | 1,680° |
| Naval brass, rolled, . | 58,700 | 18·8 | 54,500 | 17·6 | 52,600 | 16·6 | 52,000 | 16·0 | 52,000 | 14·0 | 1,640° |
| Phosphor bronze, wheel brand, cast, | 39,000 | 17·8 | 37,000 | 15·0 | 29,400 | 7·0 | 27,000 | 5·0 | 27,000 | 5·0 | 1,900° |
| Phosphor bronze, wheel brand, rolled, . . | 68,250 | 6·8 | 68,000 | 7·0 | 67,750 | 7 5 | 64,500 | 8·1 | 60,000 | 9·0 | 1,900° |
| Parsons' immadium, cast, . . | 89,600 | 20·0 | ... | ... | ... | ... | 56,400 | 25·0 | 51,500 | 22·5 | 1,660° |
| Aich's metal, rolled, | 55,250 | ... | 47,000 | ... | 42,300 | ... | 39,750 | ... | 37,000 | ... | 1,650° |
| Bull's metal, ,, | 70,000 | 16·0 | 63,600 | 13·6 | 58,700 | 13·0 | 56,750 | 14·6 | ... | ... | 1,800° |
| Aluminium, ,, | 25,300 | ... | 20,250 | ... | 15,500 | ... | 13,300 | ... | 11,500 | ... | *1,160° |
| Melloid, ,, | 98,000 | 12·0 | ... | ... | ... | ... | 90,700 | 15·0 | 87,300 | 15·0 | ... |

* Plastic State.

*Note.*—Copper at 500° has an ultimate tensile of 10·8 tons, with 40 per cent. extension.
Immadium at 500° ,, ,, 23·5 ,, 22·5 ,, ,,
Copper at 650° ,, ,, 9·7 ,, 33·5 ,, ,,

## MELTING POINTS OF METALS.

Table CXLIV. gives the melting points of the more ordinary metals as determined by the latest researches.

### Table CXLIV.—Melting Points of Metals, etc.

| Materials. | Degrees. | | Materials. | Degrees. | |
|---|---|---|---|---|---|
| | Centi. | Fahr. | | Centi. | Fahr. |
| Water . . | 0 | 32 | Pig iron (white) . | 1135 | 2075 |
| Mercury . . | –39 | –39 | ,, (grey) . | 1121 | 2050 |
| Paraffin Wax . | 54 | 129 | ,, (forge) . | 1220 | 2192 |
| Sulphur . . | 115 | 239 | Iron, grey, second cast . . . | 1240 | 2264 |
| Tin . . . | 235 | 455 | Nickel . . | 1420 | 2588 |
| Bismuth . . | 270 | 518 | Steel, high carbon | 1315 | 2400 |
| Lead . . . | 330 | 630 | ,, low ,, | 1580 | 2876 |
| Zinc . . . | 415 | 779 | Wrought iron welds | 1300 | 2550 |
| Antimony . . | 621 | 1150 | ,, ,, melts | 1600 | 2910 |
| Aluminium . | 625 | 1157 | Pure iron . . | 1525 | 2769 |
| Brass. . . | 900 | 1650 | Platinum . . | 1775 | 3217 |
| Bronze or gun-metal . . | 905 | 1660 | Heat, dull red . | 700 | 1290 |
| Bronze phosphor | 1038 | 1900 | ,, cherry-red . | 800 | 1470 |
| Silver . . | 954 | 1750 | ,, orange . | 1140 | 2100 |
| Copper . . | 1054 | 1929 | ,, white. . | 1320 | 2370 |
| Cobalt . . | 1467 | 2672 | ,, dazzling . | 1500 | 2730 |

### Table CXLIVa.—Some Alloys which Melt at Low Temperatures.

| Designation. | Composition. | | | | Melts at | |
|---|---|---|---|---|---|---|
| | Bismuth. | Lead. | Tin. | Cadmium. | C.° | F.° |
| B.C.L.T. No. 1 . | 50·0 | 25·0 | 12·5 | 12·5 | 65 | 149 |
| ,, ,, 2 . | 50·1 | 26·6 | 13·3 | 10·0 | 70 | 158 |
| ,, ,, 3 . | 27·5 | 27·5 | 10·0 | 34·5 | 75 | 167 |
| ,, ,, 4 . | ... | 25·0 | 50·0 | 25·0 | 86 | 187 |
| B.L.T. No. 1 . | 50·0 | 31·2 | 18·7 | ... | 94 | 201 |
| ,, ,, 2 . | 40·0 | 40·0 | 20·0 | ... | 113 | 235 |
| ,, ,, 3 . | 30·8 | 38·4 | 30·8 | ... | 130 | 266 |
| ,, ,, 4 . | 25·0 | 50·0 | 25·0 | ... | 149 | 300 |
| ,, ,, 5 . | 16·0 | 36·0 | 48·0 | ... | 155 | 311 |
| ,, ,, 6 . | 13·3 | 46·6 | 40·1 | ... | 165 | 329 |
| ,, ,, 7 . | 12·5 | 50·0 | 37·5 | ... | 178 | 352 |

## Table CXLIVb.—Specific Heat of various Metals, &c., Water being Unity.

| Material. | Specific Heat. | Material. | Specific Heat. |
|---|---|---|---|
| Aluminium, | 0·234 | Oxygen (equal weight), | 0·156 |
| Antimony, | 0·051 | Steam (saturated) (equal weight), | 0·305 |
| Bismuth, | 0·031 | Steam (superheated) (equal weight), | 0·370 |
| Brass (condenser tubes, etc.), | 0·094 | Alcohol, | 0·659 |
| Copper, | 0·095 | Ether, | 0·521 |
| Gold, | 0·032 | Mercury, | 0·033 |
| Iron (castings), | 0·130 | Olive oil, | 0·310 |
| Iron and steel, wrought, | 0·110 | Petroleum, | 0·434 |
| Lead, | 0·031 | Petrol, | 0·303 |
| Manganese, | 0·144 | Turpentine, | 0·416 |
| Nickel, | 0·109 | Water at 212° F. (=15 lbs. absolute pressure), | 1·0130 |
| Platinum, | 0·033 | Water at 307° F. (=75 lbs. absolute pressure), | 1·0270 |
| Silver, | 0·057 | Water at 328° F. (=100 lbs. absolute pressure), | 1·0308 |
| Steel (high carbon), | 0·117 | Water at 344° F. (=125 lbs. absolute pressure), | 1·0338 |
| Tin, | 0·057 | Water at 358° F. (=150 lbs. absolute pressure), | 1·0368 |
| Tungsten, | 0·036 | Water at 371° F. (=175 lbs. absolute pressure), | 1·0393 |
| Zinc, | 0·096 | Water at 381° F. (=200 lbs. absolute pressure), | 1·0415 |
| Brick work, | 0·192 | Water at 391° F. (=225 lbs. absolute pressure), | 1·0436 |
| Coal (Welsh), | 0·201 | Water at 401° F. (=250 lbs. absolute pressure), | 1·0460 |
| ,, Bituminous, | 0·241 | Water at 409° F. (=275 lbs. absolute pressure), | 1·0480 |
| Coke, | 0·203 | Water at 417° F. (=300 lbs. absolute pressure), | 1·0500 |
| Glass, | 0·198 | | |
| Graphite, | 0·202 | | |
| Marble and limestone, | 0·217 | | |
| Air at constant pressure (equal weight), | 0·238 | | |
| Air at constant volume (equal weight), | 0·169 | | |
| Carbonic acid (Co$_2$) (equal weight), | 0·171 | | |
| Carburetted hydrogen (equal weight), | 0·468 | | |
| Nitrogen (equal weight), | 0·174 | | |

## Table CXLIVc.—Thermal Conductivity of Metals at 18° to 20° C.

| | | | |
|---|---|---|---|
| Silver | 100·0 | Cobalt | 14·7 |
| Copper | 91·8 | Nickel | 14·0 |
| Gold | 53·2 | Iron | 11·9 |
| Aluminium | 31·3 | Lead | 8·5 |
| Zinc | 28·1 | Platinum | 8·4 |
| Tin | 15·2 | Bismuth | 1·8 |

## Table CXLIVd.—Electrical Resistance.

| Materials. | Resistance in Microhms at 0° C. per cubic inch. |
|---|---|
| Silver, annealed | 0·5874 |
| Copper, ,, | 0·6378 |
| ,, hard drawn | 0·6382 |
| Gold, ,, | 0·8177 |
| Aluminium, annealed | 1·1370 |
| ,, hard drawn | ... |
| Zinc, ,, | 2·1970 |
| Cobalt | 3·189 |
| Platinum, annealed | 3·5360 |
| Iron ,, | 3·7940 |
| Nickel ,, | 4·8660 |
| Tin, pressed | 5·1570 |
| Lead ,, | 7·6650 |
| Antimony, ,, | 13·8600 |
| Bismuth, ,, | 51·2200 |
| Mercury, liquid | 37·1400 |
| German silver | 11·8000 |
| Manganese copper (70Cu + 30Mn) | 40·5000 |
| Nickel manganese copper (73Cu + 24Mn + 3Ni) | 18·8000 |

*Note.*—The recording resistance pyrometer is now so perfect that differences of one-fifth of 1° F. in the neighbourhood of, say, 2000° F. can be measured. But the results obtained as to the melting points of metals by different observers still differ by some 30° to 50° F., partly owing to slight differences in the conditions, and partly owing to the fact that no metal solidifies entirely at one temperature. In the case of pure metals the "freezing range" is small, but in that of alloys it is often considerable, and the "freezing curve" shows several distinct freezing points.

In the case of bronze containing 90 per cent. copper and 10 per cent. tin, for instance, there are two well-marked freezing points,—one at about 1800° F. and another at about 1470° F.,—the "freezing range" being thus about 330° F.

For further information see Reports of Alloys Research Committee of Inst. of Mech. Engineers.

## WEIGHTS OF MATERIALS.

### Table CXLV.—Weights of Materials (Summary).

| Material. | Weight of a cubic foot in lbs. | Weight of a cubic inch in lbs. |
|---|---|---|
| Air (32° F. and 14·7 lbs. pressure) | ·0807 | ... |
| Water, pure, | 62·4 | ·036 |
| river, | 63 | ... |
| sea, | 64 | ·037 |
| Colza, linseed, or olive oil, | 57–58 | ... |
| Mineral oils, | 56 | ... |
| Tallow, | 58 | ... |
| Waste (moderately pressed), | 11 | ... |
| Elm, pine, or fir timber,* | 30–40 | ... |
| Beech, ash, or birch, ,, | 40–45 | ... |
| Oak or teak, ,, | 45–55 | ... |
| Greenheart, ,, | about 65 | ... |
| Lignum-vitæ, ,, | about 80 | ... |
| Fire brick, | about 140 | ... |
| Wrought-iron bars or plates, | 485 | ·281 |
| Staffordshire plates, | 480 | ·277 |
| Iron forgings (large), | 477 | ·276 |
| Steel bars and plates, | 490 | ·284 |
| Steel forgings (large), | 487 | ·282 |
| Steel—Whitworth compressed, | 496 | ·287 |
| Cast-iron, | 450 | ·260 |
| Cast-steel (mild), | 490 | ·284 |
| Sheet copper, | 550 | ·318 |
| Gun-metal, | 545 | ·315 |
| Muntz-metal, | 512 | ·296 |
| Naval brass (rolled), | 530 | ·307 |
| White metal (Babbit's), | 456 | ·263 |
| Zinc cast, | 435 | ·252 |
| rolled, | 450 | ·260 |
| Lead cast, | 708 | ·408 |
| sheet, | 711 | ·411 |

| | | | | |
|---|---|---|---|---|
| A plate of | cast-iron | 1 foot square and | 1 inch thick weighs | 37·5 lbs. |
| ,, | wrought-iron | ,, | ,, | 40 ,, |
| ,, | cast-steel (mild) | ,, | ,, | 41 ,, |
| ,, | wrought-steel (mild) | ,, | ,, | 41 ,, |
| ,, | gun-metal | ,, | ,, | 45 ,, |
| ,, | rolled brass | ,, | ,, | 44 ,, |

The Admiralty reckon 40 cubic feet of bunker space as 1 ton, but the usual allowance is 45 cubic feet to the ton; the actual average bulk of a ton is about 43 cubic feet, but if taken at 45 cubic feet a fair allowance is made for the upper portions under the deck-beams, which cannot be filled.

---

* The following are the exact weights of the various kinds of pine wood in general use, as derived from a series of careful experiments made by Mr Seaton:—

| | | |
|---|---|---|
| Yellow pine, very dry, | 24 | lbs. per cubic foot. |
| ,, in planks, seasoned, | 28 | ,, ,, |
| Baltic red pine, ,, ,, | 30·5 | ,, ,, |
| Oregon pine, ,, ,, | 35 | ,, ,, |
| Kauri pine, ,, ,, | 42 | ,, ,, |
| Pitch pine, ,, ,, | 45 | ,, ,, |

## Table CXLVI.—Table of the Weight of Round and Square Wrought-Iron Bars in lbs. per lineal foot.

| Dia. or Side. | Weight in lbs. | | Dia. or Side. | Weight in lbs. | | Dia. or Side. | Weight in lbs. | |
|---|---|---|---|---|---|---|---|---|
| | Round. | Square. | | Round. | Square. | | Round. | Square. |
| 3/16 | ·093 | ·117 | 3¾ | 36·812 | 46·875 | 8 | 167·53 | 213·33 |
| ¼ | ·164 | ·208 | ⅞ | 39·306 | 50·052 | ⅛ | 172·81 | 220·05 |
| 5/16 | ·256 | ·326 | | | | ¼ | 178·17 | 226·88 |
| ⅜ | ·368 | ·469 | 4 | 41·884 | 53·333 | ⅜ | 183·61 | 233·80 |
| 7/16 | ·501 | ·638 | ⅛ | 44·542 | 56·719 | ½ | 189·13 | 240·83 |
| ½ | ·654 | ·833 | ¼ | 47·283 | 60·208 | ⅝ | 194·73 | 247·97 |
| 9/16 | ·828 | 1·055 | ⅜ | 50·105 | 63·802 | ¾ | 200·42 | 255·21 |
| ⅝ | 1·023 | 1·302 | ½ | 53·009 | 67·500 | ⅞ | 206·19 | 262·55 |
| 11/16 | 1·237 | 1·576 | ⅝ | 55·995 | 71·302 | | | |
| ¾ | 1·473 | 1·875 | ¾ | 59·062 | 75·208 | 9 | 212·04 | 270·00 |
| 13/16 | 1·728 | 2·201 | ⅞ | 62·212 | 79·219 | ⅛ | 217·97 | 277·55 |
| ⅞ | 2·004 | 2·552 | | | | ¼ | 223·98 | 285·21 |
| 15/16 | 2·300 | 2·930 | 5 | 65·443 | 83·333 | ⅜ | 230·07 | 292·97 |
| | | | ⅛ | 68·756 | 87·552 | ½ | 236·25 | 300·83 |
| 1 | 2·618 | 3·333 | ¼ | 72·151 | 91·875 | ⅝ | 242·51 | 308·80 |
| ⅛ | 3·313 | 4·219 | ⅜ | 75·628 | 96·302 | ¾ | 248·85 | 316·88 |
| ¼ | 4·090 | 5·208 | ½ | 79·186 | 100·83 | ⅞ | 255·27 | 325·05 |
| ⅜ | 4·949 | 6·302 | ⅝ | 82·827 | 105·47 | | | |
| ½ | 5·890 | 7·500 | ¾ | 86·549 | 110·21 | 10 | 261·77 | 333·33 |
| ⅝ | 6·912 | 8·802 | ⅞ | 90·353 | 115·05 | ⅛ | 268·36 | 341·72 |
| ¾ | 8·017 | 10·208 | | | | ¼ | 275·03 | 350·21 |
| ⅞ | 9·203 | 11·719 | 6 | 94·238 | 120·00 | ⅜ | 281·77 | 358·80 |
| | | | ⅛ | 98·206 | 125·05 | ½ | 288·60 | 367·50 |
| 2 | 10·471 | 13·333 | ¼ | 102·26 | 130·21 | ⅝ | 295·52 | 376·30 |
| ⅛ | 11·821 | 15·052 | ⅜ | 106·39 | 135·47 | ¾ | 302·51 | 385·21 |
| ¼ | 13·252 | 16·875 | ½ | 110·60 | 140·83 | ⅞ | 309·59 | 394·22 |
| ⅜ | 14·766 | 18·802 | ⅝ | 114·89 | 146·30 | | | |
| ½ | 16·361 | 20·833 | ¾ | 119·27 | 151·88 | 11 | 316·75 | 403·33 |
| ⅝ | 18·038 | 22·969 | ⅞ | 123·73 | 157·55 | ⅛ | 323·99 | 412·55 |
| ¾ | 19·797 | 25·208 | | | | ¼ | 331·31 | 421·88 |
| ⅞ | 21·637 | 27·552 | 7 | 128·27 | 163·33 | ⅜ | 338·71 | 431·30 |
| | | | ⅛ | 132·89 | 169·22 | ½ | 346·20 | 440·83 |
| 3 | 23·560 | 30·000 | ¼ | 137·60 | 175·21 | ⅝ | 353·76 | 450·47 |
| ⅛ | 25·564 | 32·552 | ⅜ | 142·98 | 181·30 | ¾ | 361·41 | 460·21 |
| ¼ | 27·650 | 35·208 | ½ | 147·25 | 187·50 | ⅞ | 369·14 | 470·05 |
| ⅜ | 29·818 | 37·969 | ⅝ | 152·20 | 193·80 | | | |
| ½ | 32·067 | 40·833 | ¾ | 157·23 | 200·21 | 12 | 376·95 | 480·00 |
| ⅝ | 34·399 | 43·802 | ⅞ | 162·34 | 206·72 | | | |

**For larger sizes take weight of bar of half the diameter or side and multiply it by four** (*vide* page 481).

## Table CXLVII.—Weight of Round and Square Bar Steel in lbs. per lineal foot.

| Dia. or Side. | Weight in lbs. | | Dia. or Side. | Weight in lbs. | | Dia. or Side. | Weight in lbs. | |
|---|---|---|---|---|---|---|---|---|
| | Round. | Square. | | Round. | Square. | | Round. | Square. |
| ⅛ | ·042 | ·053 | 3⅝ | 35·090 | 44·678 | 7⅞ | 165·60 | 210·85 |
| 3/16 | ·094 | ·120 | ¾ | 37·552 | 47·813 | | | |
| ¼ | ·167 | ·213 | ⅞ | 40·097 | 51·053 | 8 | 170·90 | 217·60 |
| 5/16 | ·261 | ·332 | | | | ⅛ | 176·29 | 225·25 |
| ⅜ | ·375 | ·478 | 4 | 42·726 | 54·400 | ¼ | 181·75 | 231·41 |
| 7/16 | ·511 | ·651 | ⅛ | 45·438 | 57·853 | ⅜ | 187·30 | 238·48 |
| ½ | ·667 | ·850 | ¼ | 48·233 | 61·413 | ½ | 192·93 | 245·65 |
| 9/16 | ·845 | 1·076 | ⅜ | 51·112 | 65·078 | ⅝ | 198·65 | 252·93 |
| ⅝ | 1·043 | 1·328 | ½ | 54·075 | 68·850 | ¾ | 204·45 | 260·31 |
| 11/16 | 1·262 | 1·607 | ⅝ | 57·121 | 72·728 | ⅞ | 210·33 | 267·80 |
| ¾ | 1·502 | 1·913 | ¾ | 60·250 | 76·713 | | | |
| 13/16 | 1·762 | 2·245 | ⅞ | 63·463 | 80·803 | 9 | 216·30 | 275·40 |
| ⅞ | 2·044 | 2·603 | | | | ⅛ | 222·35 | 283·10 |
| 15/16 | 2·347 | 2·988 | 5 | 66·759 | 85·000 | ¼ | 228·48 | 290·91 |
| | | | ⅛ | 70·139 | 89·303 | ⅜ | 234·70 | 298·83 |
| 1 | 2·670 | 3·400 | ¼ | 73·602 | 93·713 | ½ | 241·00 | 306·85 |
| ⅛ | 3·380 | 4·303 | ⅜ | 77·148 | 98·229 | ⅝ | 248·38 | 314·98 |
| ¼ | 4·172 | 5·313 | ½ | 80·778 | 102·85 | ¾ | 253·85 | 323·21 |
| ⅜ | 5·049 | 6·428 | ⅝ | 84·492 | 107·58 | ⅞ | 260·40 | 331·55 |
| ½ | 6·008 | 7·650 | ¾ | 88·288 | 112·41 | | | |
| ⅝ | 7·051 | 8·978 | ⅞ | 92·169 | 117·35 | 10 | 267·04 | 340·00 |
| ¾ | 8·178 | 10·413 | | | | ⅛ | 273·75 | 348·55 |
| ⅞ | 9·388 | 11·953 | 6 | 96·133 | 122·40 | ¼ | 280·55 | 357·21 |
| | | | ⅛ | 100·18 | 127·55 | ⅜ | 287·44 | 365·98 |
| 2 | 10·681 | 13·600 | ¼ | 104·31 | 132·81 | ½ | 294·41 | 374·85 |
| ⅛ | 12·058 | 15·353 | ⅜ | 108·52 | 138·18 | ⅝ | 301·46 | 383·83 |
| ¼ | 13·519 | 17·213 | ½ | 112·82 | 143·65 | ¾ | 308·59 | 392·91 |
| ⅜ | 15·062 | 19·178 | ⅝ | 117·20 | 149·23 | ⅞ | 315·81 | 402·10 |
| ½ | 16·690 | 21·250 | ¾ | 121·67 | 154·91 | | | |
| ⅝ | 18·400 | 23·428 | ⅞ | 126·22 | 160·70 | 11 | 323·11 | 411·40 |
| ¾ | 20·195 | 25·713 | | | | ⅛ | 330·50 | 420·80 |
| ⅞ | 22·072 | 28·103 | 7 | 130·85 | 166·60 | ¼ | 337·97 | 430·31 |
| | | | ⅛ | 135·56 | 172·60 | ⅜ | 345·52 | 439·93 |
| 3 | 24·033 | 30·600 | ¼ | 140·36 | 178·71 | ½ | 353·15 | 449·65 |
| ⅛ | 26·078 | 33·203 | ⅜ | 145·24 | 184·93 | ⅝ | 360·87 | 459·48 |
| ¼ | 28·206 | 35·913 | ½ | 150·21 | 191·25 | ¾ | 368·68 | 469·41 |
| ⅜ | 30·417 | 38·728 | ⅝ | 155·26 | 197·68 | ⅞ | 376·56 | 479·45 |
| ½ | 32·712 | 41·650 | ¾ | 160·39 | 204·21 | 12 | 384·53 | 489·60 |

**For larger sizes take weight of bar of half the diameter or side and multiply it by four** (*vide* page 482).

## Table CXLVIIa.—Round Steel Shafts, Weight per foot in lbs.

| Diameter. | Weight. | Diameter. | Weight. | Diameter. | Weight. |
|---|---|---|---|---|---|
| 12¼ | 400·7 | 16¼ | 705·2 | 20¼ | 1095 |
| 12½ | 417·2 | 16½ | 727·0 | 20½ | 1122 |
| 12¾ | 434·1 | 16¾ | 749·2 | 20¾ | 1150 |
| 13 | 451·3 | 17 | 771·7 | 21 | 1178 |
| 13¼ | 468·8 | 17¼ | 794·6 | 21¼ | 1206 |
| 13½ | 486·7 | 17½ | 817·8 | 21½ | 1234 |
| 13¾ | 504·9 | 17¾ | 841·3 | 21¾ | 1263 |
| 14 | 523·4 | 18 | 865·2 | 22 | 1292 |
| 14¼ | 542·2 | 18¼ | 889·4 | 22¼ | 1322 |
| 14½ | 561·4 | 18½ | 913·9 | 22½ | 1352 |
| 14¾ | 581·0 | 18¾ | 938·8 | 22¾ | 1382 |
| 15 | 600·8 | 19 | 964·0 | 23 | 1413 |
| 15¼ | 621·0 | 19¼ | 993·5 | 23¼ | 1444 |
| 15½ | 641·6 | 19½ | 1015 | 23½ | 1475 |
| 15¾ | 662·4 | 19¾ | 1042 | 22¾ | 1506 |
| 16 | 683·6 | 20 | 1070 | 24 | 1538 |

## Table CXLVIIb.—Round Hollow Shafts—Bore, Half of Outside Diameter.

| Diameter. | Weight. | Diameter. | Weight. | Diameter. | Weight. |
|---|---|---|---|---|---|
| ins. | | ins. | | ins. | |
| 10 | 200 | 14½ | 421 | 19 | 723 |
| 10½ | 221 | 15 | 450 | 19½ | 761 |
| 11 | 242 | 15½ | 480 | 20 | 802 |
| 11½ | 265 | 16 | 513 | 20½ | 841 |
| 12 | 288 | 16½ | 545 | 21 | 883 |
| 12½ | 313 | 17 | 579 | 21½ | 925 |
| 13 | 338 | 17½ | 613 | 22 | 969 |
| 13½ | 364 | 18 | 649 | 23 | 1059 |
| 14 | 391 | 18½ | 686 | 24 | 1153 |

Table CXLVIII.—Weight of Flat Wrought-Iron Bars in lbs. per lineal foot.

| Breadth of bar (ins.). | Thickness in fractions of an inch. | | | | | | | | | | | | | Breadth of bar (ins.). |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | ¼ | 5/16 | ⅜ | 7/16 | ½ | 9/16 | ⅝ | 1/16 | ¾ | 13/16 | ⅞ | 15/16 | 1 in. | |
| 1 | ·83 | 1·04 | 1·25 | 1·46 | 1·67 | 1·88 | 2·08 | 2·29 | 2·50 | 2·71 | 2·92 | 3·13 | 3·33 | 1 |
| 1¼ | 1·04 | 1·30 | 1·56 | 1·82 | 2·08 | 2·34 | 2·60 | 2·86 | 3·13 | 3·39 | 3·65 | 3·91 | 4·16 | 1¼ |
| 1½ | 1·25 | 1·56 | 1·88 | 2·19 | 2·50 | 2·81 | 3·13 | 3·44 | 3·75 | 4·06 | 4·38 | 4·69 | 5·00 | 1½ |
| 1¾ | 1·46 | 1·82 | 2·19 | 2·55 | 2·92 | 3·28 | 3·65 | 4·01 | 4·38 | 4·74 | 5·10 | 5·47 | 5·83 | 1¾ |
| 2 | 1·67 | 2·08 | 2·50 | 2·92 | 3·33 | 3·75 | 4·17 | 4·58 | 5·00 | 5·42 | 5·83 | 6·25 | 6·67 | 2 |
| 2¼ | 1·88 | 2·34 | 2·81 | 3·28 | 3·75 | 4·22 | 4·69 | 5·16 | 5·63 | 6·09 | 6·56 | 7·03 | 7·50 | 2¼ |
| 2½ | 2·08 | 2·60 | 3·13 | 3·65 | 4·17 | 4·69 | 5·21 | 5·73 | 6·25 | 6·77 | 7·29 | 7·81 | 8·33 | 2½ |
| 2¾ | 2·29 | 2·86 | 3·44 | 4·01 | 4·58 | 5·16 | 5·73 | 6·30 | 6·88 | 7·45 | 8·02 | 8·59 | 9·17 | 2¾ |
| 3 | 2·50 | 3·13 | 3·75 | 4·38 | 5·00 | 5·63 | 6·25 | 6·88 | 7·50 | 8·13 | 8·75 | 9·38 | 10·00 | 3 |
| 3¼ | 2·71 | 3·39 | 4·06 | 4·74 | 5·42 | 6·09 | 6·77 | 7·45 | 8·13 | 8·80 | 9·48 | 10·16 | 10·83 | 3¼ |
| 3½ | 2·92 | 3·65 | 4·38 | 5·10 | 5·83 | 6·56 | 7·29 | 8·02 | 8·75 | 9·48 | 10·21 | 10·94 | 11·67 | 3½ |
| 3¾ | 3·13 | 3·91 | 4·69 | 5·47 | 6·25 | 7·03 | 7·81 | 8·59 | 9·38 | 10·16 | 10·94 | 11·72 | 12·50 | 3¾ |
| 4 | 3·33 | 4·17 | 5·00 | 5·83 | 6·67 | 7·50 | 8·33 | 9·17 | 10·00 | 10·83 | 11·67 | 12·50 | 13·33 | 4 |
| 4¼ | 3·54 | 4·43 | 5·31 | 6·20 | 7·08 | 7·97 | 8·85 | 9·74 | 10·63 | 11·51 | 12·40 | 13·28 | 14·17 | 4¼ |
| 4½ | 3·75 | 4·69 | 5·63 | 6·56 | 7·50 | 8·44 | 9·38 | 10·31 | 11·25 | 12·19 | 13·13 | 14·06 | 15·00 | 4½ |
| 4¾ | 3·96 | 4·95 | 5·94 | 6·93 | 7·92 | 8·91 | 9·90 | 10·89 | 11·88 | 12·86 | 13·85 | 14·84 | 15·83 | 4¾ |
| 5 | 4·17 | 5·21 | 6·25 | 7·29 | 8·33 | 9·38 | 10·42 | 11·46 | 12·50 | 13·54 | 14·58 | 15·63 | 16·67 | 5 |

Table CXLIX.—Weight of Flat Rolled Steel Bars in lbs. per lineal foot.

| Breadth of bar (ins.). | Thickness in fractions of an inch. | | | | | | | | | | | | | Breadth of bar (ins.). |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | ¼ | 5/16 | ⅜ | 7/16 | ½ | 9/16 | ⅝ | 11/16 | ¾ | 13/16 | ⅞ | 15/16 | 1 in. | |
| 1 | ·85 | 1·06 | 1·28 | 1·49 | 1·70 | 1·91 | 2·13 | 2·34 | 2·55 | 2·76 | 2·98 | 3·19 | 3·40 | 1 |
| 1¼ | 1·06 | 1·33 | 1·59 | 1·86 | 2·13 | 2·39 | 2·66 | 2·92 | 3·19 | 3·45 | 3·72 | 3·98 | 4·25 | 1¼ |
| 1½ | 1·28 | 1·59 | 1·91 | 2·23 | 2·55 | 2·87 | 3·19 | 3·51 | 3·83 | 4·14 | 4·46 | 4·78 | 5·10 | 1½ |
| 1¾ | 1·49 | 1·86 | 2·23 | 2·60 | 2·98 | 3·35 | 3·72 | 4·09 | 4·46 | 4·83 | 5·21 | 5·58 | 5·95 | 1¾ |
| 2 | 1·70 | 2·13 | 2·55 | 2·98 | 3·40 | 3·83 | 4·25 | 4·68 | 5·10 | 5·53 | 5·95 | 6·38 | 6·80 | 2 |
| 2¼ | 1·91 | 2·39 | 2·87 | 3·35 | 3·82 | 4·30 | 4·78 | 5·26 | 5·74 | 6·22 | 6·69 | 7·17 | 7·65 | 2¼ |
| 2½ | 2·13 | 2·66 | 3·19 | 3·72 | 4·25 | 4·78 | 5·31 | 5·84 | 6·38 | 6·91 | 7·44 | 7·97 | 8·50 | 2½ |
| 2¾ | 2·34 | 2·92 | 3·51 | 4·09 | 4·68 | 5·26 | 5·84 | 6·43 | 7·01 | 7·60 | 8·18 | 8·77 | 9·35 | 2¾ |
| 3 | 2·55 | 3·19 | 3·83 | 4·46 | 5·10 | 5·74 | 6·38 | 7·01 | 7·65 | 8·29 | 8·93 | 9·56 | 10·20 | 3 |
| 3¼ | 2·76 | 3·45 | 4·14 | 4·83 | 5·53 | 6·22 | 6·91 | 7·60 | 8·29 | 8·98 | 9·67 | 10·36 | 11·05 | 3¼ |
| 3½ | 2·98 | 3·72 | 4·46 | 5·21 | 5·95 | 6·69 | 7·44 | 8·18 | 8·93 | 9·67 | 10·41 | 11·16 | 11·90 | 3½ |
| 3¾ | 3·19 | 3·98 | 4·78 | 5·58 | 6·38 | 7·17 | 7·97 | 8·77 | 9·56 | 10·36 | 11·16 | 11·95 | 12·75 | 3¾ |
| 4 | 3·40 | 4·25 | 5·10 | 5·95 | 6·80 | 7·65 | 8·50 | 9·35 | 10·20 | 11·05 | 11·90 | 12·75 | 13·60 | 4 |
| 4¼ | 3·61 | 4·52 | 5·42 | 6·32 | 7·22 | 8·13 | 9·03 | 9·93 | 10·84 | 11·74 | 12·64 | 13·55 | 14·45 | 4¼ |
| 4½ | 3·83 | 4·78 | 5·74 | 6·69 | 7·65 | 8·61 | 9·56 | 10·52 | 11·48 | 12·43 | 13·39 | 14·34 | 15·30 | 4½ |
| 4¾ | 4·04 | 5·05 | 6·06 | 7·07 | 8·08 | 9·08 | 10·09 | 11·10 | 12·11 | 13·12 | 14·13 | 15·14 | 16·15 | 4¾ |
| 5 | 4·25 | 5·31 | 6·38 | 7·44 | 8·50 | 9·56 | 10·63 | 11·69 | 12·75 | 13·81 | 14·88 | 15·94 | 17·00 | 5 |

## Table CXLIXa.—Weight of Rectangular Steel Bars per foot in lbs.

| Breadth of bar. | Thickness in inches. | | | | | | | | | | | | Breadth of bar. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1¼ | 1½ | 1¾ | 2 | 2¼ | 2½ | 2¾ | 3 | 3¼ | 3½ | 3¾ | 4 | |
| 6 | 25·5 | 30·60 | 35·7 | 40·8 | 45·9 | 51·0 | 56·1 | 61·2 | 66·3 | 71·4 | 76·5 | 81·6 | 6 |
| 7 | 29·75 | 35·70 | 41·7 | 47·6 | 53·6 | 59·5 | 65·5 | 71·4 | 77·4 | 83·3 | 89·3 | 95·2 | 7 |
| 8 | 34·00 | 40·80 | 47·7 | 54·4 | 61·2 | 68·0 | 74·8 | 81·6 | 88·4 | 95·2 | 102·0 | 108·8 | 8 |
| 9 | 38·25 | 45·9 | 53·6 | 61·2 | 68·9 | 76·5 | 84·2 | 91·8 | 99·5 | 107·1 | 114·8 | 122·4 | 9 |
| 10 | 42·50 | 51·0 | 59·5 | 68·0 | 76·5 | 85·0 | 93·5 | 102·0 | 110·5 | 119·0 | 127·5 | 136·0 | 10 |
| 11 | 46·75 | 56·1 | 65·5 | 74·8 | 84·2 | 93·5 | 102·9 | 112·2 | 121·6 | 130·9 | 140·3 | 149·6 | 11 |
| 12 | 51·00 | 61·2 | 71·4 | 81·6 | 91·8 | 102·0 | 112·2 | 122·4 | 132·6 | 142·8 | 153·0 | 163·2 | 12 |
| 13 | 55·25 | 66·3 | 77·4 | 88·4 | 99·5 | 110·5 | 121·6 | 132·6 | 143·7 | 154·7 | 165·8 | 176·8 | 13 |
| 14 | 59·50 | 71·4 | 83·3 | 95·2 | 107·1 | 119·0 | 130·9 | 142·8 | 154·7 | 166·6 | 178·5 | 190·4 | 14 |
| 15 | 63·75 | 76·5 | 89·3 | 102·0 | 114·8 | 127·5 | 140·3 | 153·0 | 165·8 | 178·5 | 191·3 | 204·0 | 15 |
| 16 | 68·00 | 81·6 | 95·2 | 108·8 | 122·4 | 136·0 | 149·6 | 163·2 | 176·8 | 190·4 | 204·0 | 217·6 | 16 |

## Table CL.—Weight of Iron Angle-Bars in lbs. per lineal foot.

| Sum of flanges (ins.). | Thickness in fractions of an inch. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3/16 | ¼ | 5/16 | ⅜ | 7/16 | ½ | 9/16 | ⅝ | 11/16 | ¾ | 13/16 |
| 2½ | 1·45 | 1·88 | 2·28 | 2·66 | ... | ... | ... | ... | ... | ... | ... |
| 2¾ | 1·60 | 2·08 | 2·54 | 2·97 | ... | ... | ... | ... | ... | ... | ... |
| 3 | 1·76 | 2·29 | 2·80 | 3·28 | ... | ... | ... | ... | ... | ... | ... |
| 3¼ | 1·91 | 2·50 | 3·06 | 3·59 | 4·10 | ... | ... | ... | ... | ... | ... |
| 3½ | 2·07 | 2·71 | 3·32 | 3·91 | 4·47 | ... | ... | ... | ... | ... | ... |
| 3¾ | 2·23 | 2·92 | 3·58 | 4·22 | 4·83 | ... | ... | ... | ... | ... | ... |
| 4 | 2·38 | 3·13 | 3·84 | 4·53 | 5·20 | 5·83 | ... | ... | ... | ... | ... |
| 4¼ | 2·54 | 3·33 | 4·10 | 4·84 | 5·56 | 6·25 | ... | ... | ... | ... | ... |
| 4½ | 2·70 | 3·54 | 4·36 | 5·16 | 5·92 | 6·67 | ... | ... | ... | ... | ... |
| 4¾ | 2·85 | 3·75 | 4·62 | 5·47 | 6·29 | 7·08 | 7·85 | ... | ... | ... | ... |
| 5 | ... | 3·96 | 4·88 | 5·78 | 6·65 | 7·50 | 8·32 | ... | ... | ... | ... |
| 5¼ | ... | 4·17 | 5·14 | 6·09 | 7·02 | 7·92 | 8·79 | ... | ... | ... | ... |
| 5½ | ... | 4·38 | 5·40 | 6·41 | 7·38 | 8·33 | 9·26 | 10·16 | ... | ... | ... |
| 5¾ | ... | ... | 5·66 | 6·72 | 7·75 | 8·75 | 9·73 | 10·68 | ... | ... | ... |
| 6 | ... | ... | 5·92 | 7·03 | 8·11 | 9·17 | 10·20 | 11·20 | ... | ... | ... |
| 6¼ | ... | ... | 6·18 | 7·34 | 8·48 | 9·58 | 10·66 | 11·72 | 12·75 | ... | ... |
| 6½ | ... | ... | ... | 7·66 | 8·84 | 10·00 | 11·13 | 12·24 | 13·32 | ... | .. |
| 6¾ | ... | ... | ... | 7·97 | 9·21 | 10·42 | 11·60 | 12·76 | 13·89 | ... | ... |
| 7 | ... | ... | ... | 8·28 | 9·57 | 10·83 | 12·07 | 13·28 | 14·46 | 15·63 | ... |
| 7¼ | ... | ... | ... | ... | 9·93 | 11·25 | 12·54 | 13·80 | 15·04 | 16·25 | ... |
| 7½ | ... | ... | ... | ... | 10·30 | 11·67 | 13·01 | 14·32 | 15·61 | 16·88 | ... |
| 7¾ | ... | ... | ... | ... | 10·66 | 12·08 | 13·48 | 14·84 | 16·18 | 17·50 | 18·79 |
| 8 | ... | ... | ... | ... | ... | 12·50 | 13·95 | 15·36 | 16·76 | 18·13 | 19·47 |
| 8¼ | ... | ... | ... | ... | ... | 12·92 | 14·44 | 15·89 | 17·33 | 18·75 | 20·14 |
| 8½ | ... | ... | ... | ... | ... | 13·33 | 14·88 | 16·41 | 17·90 | 19·38 | 20·82 |
| 8¾ | ... | ... | ... | ... | ... | ... | 15·35 | 16·93 | 18·48 | 20·00 | 21·50 |
| 9 | ... | ... | ... | ... | ... | ... | 15·82 | 17·45 | 19·05 | 20·63 | 22·17 |
| 9¼ | ... | ... | ... | ... | ... | ... | 16·29 | 17·97 | 19·62 | 21·25 | 22·85 |
| 9½ | ... | ... | ... | ... | ... | ... | ... | 18·49 | 20·20 | 21·88 | 23·53 |
| 9¾ | ... | ... | ... | ... | ... | ... | ... | 19·01 | 20·77 | 22·50 | 24·21 |
| 10 | ... | ... | ... | ... | ... | ... | ... | 19·53 | 21·34 | 23·13 | 24·88 |

## Table CLI.—Weight of Steel Angle-Bars in lbs. per lineal foot.

| Sum of flanges (ins.). | Thickness in fractions of an inch. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3/16 | ¼ | 5/16 | ⅜ | 7/16 | ½ | 9/16 | ⅝ | 11/16 | ¾ | 13/16 |
| 2½ | 1·47 | 1·91 | 2·32 | 2·71 | ... | ... | ... | ... | ... | ... | ... |
| 2¾ | 1·63 | 2·12 | 2·59 | 3·03 | ... | ... | ... | ... | .. | ... | ... |
| 3 | 1·79 | 2·34 | 2·86 | 3·35 | ... | ... | ... | ... | ... | ... | ... |
| 3¼ | 1·95 | 2·55 | 3·12 | 3·67 | 4·18 | ... | ... | ... | ... | ... | ... |
| 3½ | 2·11 | 2·76 | 3·39 | 3·98 | 4·56 | ... | ... | ... | ... | ... | ... |
| 3¾ | 2·27 | 2·97 | 3·65 | 4·30 | 4·93 | ... | ... | ... | ... | ... | ... |
| 4 | 2·43 | 3·19 | 3·92 | 4·62 | 5·30 | 5·95 | ... | ... | ... | ... | ... |
| 4¼ | 2·59 | 3·40 | 4·18 | 4·94 | 5·67 | 6·38 | ... | ... | ... | ... | ... |
| 4½ | 2·75 | 3·61 | 4·45 | 5·26 | 6·04 | 6·80 | ... | ... | ... | ... | ... |
| 4¾ | 2·91 | 3·82 | 4·71 | 5·58 | 6·41 | 7·23 | 8·01 | ... | ... | ... | ... |
| 5 | ... | 4·04 | 4·98 | 5·90 | 6·79 | 7·65 | 8·49 | ... | ... | ... | ... |
| 5¼ | ... | 4·25 | 5·25 | 6·22 | 7·16 | 8·08 | 8·96 | ... | ... | ... | ... |
| 5½ | ... | 4·46 | 5·51 | 6·53 | 7·53 | 8·50 | 9·44 | 10·36 | ... | ... | ... |
| 5¾ | ... | ... | 5·78 | 6·85 | 7·90 | 8·93 | 9·92 | 10·89 | ... | ... | ... |
| 6 | ... | ... | 6·04 | 7·17 | 8·27 | 9·35 | 10·40 | 11·42 | ... | .. | ... |
| 6¼ | ... | ... | 6·31 | 7·49 | 8·65 | 9·78 | 10·88 | 11·95 | 13·00 | ... | ... |
| 6½ | ... | ... | ... | 7·81 | 9·02 | 10·20 | 11·36 | 12·48 | 13·59 | ... | ... |
| 6¾ | ... | ... | ... | 8·13 | 9·39 | 10·63 | 11·83 | 13·02 | 14·17 | ... | ... |
| 7 | ... | ... | ... | 8·45 | 9·76 | 11·05 | 12·31 | 13·55 | 14·76 | 15·94 | ... |
| 7¼ | ... | ... | ... | ... | 10·13 | 11·48 | 12·79 | 14·08 | 15·34 | 16·58 | ... |
| 7½ | ... | ... | ... | ... | 10·51 | 11·90 | 13·27 | 14·61 | 15·92 | 17·21 | ... |
| 7¾ | ... | ... | ... | ... | 10·88 | 12·33 | 13·75 | 15·14 | 16·51 | 17·85 | 19·16 |
| 8 | ... | ... | ... | ... | ... | 12·75 | 14·22 | 15·67 | 17·09 | 18·49 | 19·85 |
| 8¼ | ... | ... | ... | ... | ... | 13·18 | 14·70 | 16·20 | 17·68 | 19·13 | 20·55 |
| 8½ | ... | ... | ... | ... | ... | 13·60 | 15·18 | 16·73 | 18·26 | 19·76 | 21·24 |
| 8¾ | ... | ... | ... | ... | ... | ... | 15·66 | 17·27 | 18·85 | 20·40 | 21·93 |
| 9 | ... | ... | ... | ... | ... | ... | 16·14 | 17·80 | 19·43 | 21·04 | 22·62 |
| 9¼ | ... | ... | ... | ... | ... | ... | 16·61 | 18·33 | 20·01 | 21·68 | 23·31 |
| 9½ | ... | ... | ... | ... | ... | ... | ... | 18·86 | 20·60 | 22·31 | 24·00 |
| 9¾ | ... | ... | ... | ... | ... | ... | ... | 19·39 | 21·18 | 22·95 | 24·69 |
| 10 | ... | ... | ... | ... | ... | ... | ... | 19·92 | 21·77 | 23·59 | 25·38 |

## Table CLII.—Weights of

| External Diam. | | | THICKNESS | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | L.S.G. | | 16 | 15 | 14 | 13 | 12 | 11 | 10 | 9 | 8 | 7 | 6 | 5 |
| | Inches. | | ·064 | ·072 | ·080 | ·092 | ·104 | ·116 | ·128 | ·144 | ·160 | ·176 | ·192 | ·215 |
| | Millimetres | | 1·626 | 1·829 | 2·032 | 2·337 | 2·642 | 2·946 | 3·251 | 3·658 | 4·064 | 4·470 | 4·877 | 5·385 |
| Ins. | Usual L.S.G. and Weight. | | WEIGHT PER | | | | | | | | | | | |
| 1 | 14 | 0·771 | 0·627 | 0·700 | 0·771 | 0·875 | 0·976 | 1·074 | 1·169 | 1·291 | 1·407 | 1·519 | 1·624 | 1·749 |
| 1⅛ | 14 | 0·875 | 0·711 | 0·794 | 0·875 | 0·995 | 1·112 | 1·226 | 1·336 | 1·479 | 1·617 | 1·749 | 1·876 | 2·027 |
| 1¼ | 13 | 1·116 | | 0·888 | 0·980 | 1·116 | 1·248 | 1·377 | 1·504 | 1·668 | 1·826 | 1·979 | 2·127 | 2·304 |
| 1⅜ | 13 | 1·236 | | 0·982 | 1·085 | 1·236 | 1·384 | 1·529 | 1·671 | 1·856 | 2·036 | 2·210 | 2·378 | 2·582 |
| 1½ | 13 | 1·356 | | 1·077 | 1·190 | 1·356 | 1·520 | 1·681 | 1·839 | 2·045 | 2·245 | 2·440 | 2·630 | 2·859 |
| 1⅝ | 13 | 1·477 | | 1·171 | 1·294 | 1·477 | 1·656 | 1·833 | 2·007 | 2·233 | 2·455 | 2·671 | 2·881 | 3·137 |
| 1¾ | 13 | 1·597 | | 1·265 | 1·399 | 1·597 | 1·793 | 1·985 | 2·174 | 2·422 | 2·664 | 2·901 | 3·132 | 3·414 |
| 1⅞ | 13 | 1·718 | | 1·359 | 1·504 | 1·718 | 1·929 | 2·137 | 2·342 | 2·610 | 2·873 | 3·131 | 3·384 | 3·602 |
| 2 | 12 | 2·065 | | | 1·608 | 1·838 | 2·065 | 2·288 | 2·509 | 2·799 | 3·083 | 3·362 | 3·635 | 3·969 |
| 2⅛ | 12 | 2·201 | | | 1·713 | 1·959 | 2·201 | 2·440 | 2·677 | 2·987 | 3·292 | 3·592 | 3·886 | 4·247 |
| 2¼ | 12 | 2·337 | | | 1·818 | 2·079 | 2·337 | 2·592 | 2·844 | 3·176 | 3·502 | 3·822 | 4·138 | 4·524 |
| 2⅜ | 11 | 2·744 | | | | 2·199 | 2·473 | 2·744 | 3·012 | 3·364 | 3·711 | 4·053 | 4·389 | 4·802 |
| 2½ | 11 | 2·896 | | | | 2·320 | 2·609 | 2·896 | 3·179 | 3·553 | 3·921 | 4·283 | 4·640 | 5·079 |
| 2⅝ | 11 | 3·048 | | | | 2·440 | 2·745 | 3·048 | 3·347 | 3·741 | 4·130 | 4·514 | 4·892 | 5·357 |
| 2¾ | 11 | 3·200 | | | | 2·561 | 2·882 | 3·200 | 3·514 | 3·930 | 4·339 | 4·744 | 5·143 | 5·634 |
| 2⅞ | 11 | 3·351 | | | | 2·681 | 3·018 | 3·351 | 3·682 | 4·118 | 4·549 | 4·974 | 5·394 | 5·912 |
| 3 | 11 | 3·503 | | | | 2·802 | 3·154 | 3·503 | 3·850 | 4·307 | 4·758 | 5·205 | 5·646 | 6·189 |
| 3⅛ | 10 | 4·017 | | | | | 3·290 | 3·655 | 4·017 | 4·495 | 4·968 | 5·435 | 5·897 | 6·467 |
| 3¼ | 10 | 4·185 | | | | | 3·426 | 3·807 | 4·185 | 4·684 | 5·177 | 5·665 | 6·148 | 6·744 |
| 3⅜ | 10 | 4·352 | | | | | 3·562 | 3·959 | 4·352 | 4·872 | 5·387 | 5·896 | 6·400 | 7·022 |
| 3½ | 10 | 4·520 | | | | | 3·698 | 4·111 | 4·520 | 5·061 | 5·596 | 6·126 | 6·651 | 7·299 |
| 3⅝ | 10 | 4·687 | | | | | 3·835 | 4·262 | 4·687 | 5·249 | 5·806 | 6·357 | 6·902 | 7·577 |
| 3¾ | 10 | 4·855 | | | | | 3·971 | 4·414 | 4·855 | 5·438 | 6·015 | 6·587 | 7·154 | 7·854 |
| 3⅞ | 9 | 5·626 | | | | | | 4·566 | 5·022 | 5·626 | 6·224 | 6·817 | 7·405 | 8·132 |
| 4 | 9 | 5·815 | | | | | | 4·718 | 5·190 | 5·815 | 6·434 | 7·048 | 7·656 | 8·410 |

This table of weights is based on the specific gravity of Iron at 7·698. Actual weights may differ slightly.

For other metals—
Steel, .. .. multiply by 1·012
Brass, .. .. „ 1·080
Copper, .. .. „ 1·135

## Iron Boiler Tubes.

| OF IRON. | | | | | | | | | | | External Diam. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 4 | 3 | 2 | 1 | 1/8" | 3/16" | 1/4" | 5/16" | 3/8" | 7/16" | 1/2" | |
| ·232 | ·252 | ·276 | ·300 | ·125 | ·187 | ·250 | ·313 | ·375 | ·437 | ·500 | |
| 5·893 | 6·401 | 7·010 | 7·620 | 3·175 | 4·726 | 6·350 | 7·937 | 9·525 | 11·112 | 12·700 | |
| FOOT IN POUNDS. | | | | | | | | | | | Ins. |
| 1·866 | 1·974 | 2·092 | 2·199 | 1·145 | 1·595 | 1·963 | 2·250 | 2·454 | 2·577 | 2·618 | 1 |
| 2·169 | 2·304 | 2·454 | 2·592 | 1·309 | 1·841 | 2·291 | 2·659 | 2·945 | 3·150 | 3·272 | 1⅛ |
| 2·473 | 2·634 | 2·815 | 2·984 | 1·473 | 2·086 | 2·618 | 3·068 | 3·436 | 3·723 | 3·927 | 1¼ |
| 2·777 | 2·963 | 3·176 | 3·377 | 1·636 | 2·332 | 2·945 | 3·477 | 3·927 | 4·295 | 4·581 | 1⅜ |
| 3·081 | 3·293 | 3·538 | 3·770 | 1·800 | 2·577 | 3·272 | 3·886 | 4·418 | 4·868 | 5·236 | 1½ |
| 3·384 | 3·623 | 3·899 | 4·163 | 1·963 | 2·822 | 3·600 | 4·295 | 4·909 | 5·440 | 5·890 | 1⅝ |
| 3·688 | 3·953 | 4·260 | 4·555 | 2·127 | 3·068 | 3·927 | 4·704 | 5·400 | 6·013 | 6·545 | 1¾ |
| 3·992 | 4·283 | 4·621 | 4·948 | 2·291 | 3·313 | 4·254 | 5·113 | 5·890 | 6·586 | 7·199 | 1⅞ |
| 4·295 | 4·613 | 4·983 | 5·341 | 2·454 | 3·559 | 4·581 | 5·522 | 6·381 | 7·159 | 7·854 | 2 |
| 4·599 | 4·943 | 5·344 | 5·733 | 2·618 | 3·804 | 4·909 | 5·931 | 6·872 | 7·731 | 8·508 | 2⅛ |
| 4·903 | 5·273 | 5·705 | 6·126 | 2·782 | 4·050 | 5·236 | 6·340 | 7·363 | 8·304 | 9·163 | 2¼ |
| 5·206 | 5·602 | 6·067 | 6·519 | 2·945 | 4·295 | 5·563 | 6·749 | 7·854 | 8·877 | 9·817 | 2⅜ |
| 5·510 | 5·932 | 6·428 | 6·911 | 3·109 | 4·541 | 5·890 | 7·159 | 8·345 | 9·449 | 10·472 | 2½ |
| 5·814 | 6·262 | 6·789 | 7·304 | 3·272 | 4·786 | 6·218 | 7·568 | 8·836 | 10·022 | 11·126 | 2⅝ |
| 6·117 | 6·592 | 7·150 | 7·697 | 3·436 | 5·031 | 6·545 | 7·977 | 9·327 | 10·595 | 11·781 | 2¾ |
| 6·421 | 6·922 | 7·512 | 8·090 | 3·600 | 5·277 | 6·872 | 8·386 | 9·818 | 11·167 | 12·435 | 2⅞ |
| 6·725 | 7·252 | 7·873 | 8·482 | 3·763 | 5·522 | 7·200 | 8·795 | 10·308 | 11·740 | 13·090 | 3 |
| 7·028 | 7·582 | 8·234 | 8·875 | 3·927 | 5·768 | 7·527 | 9·204 | 10·799 | 12·313 | 13·744 | 3⅛ |
| 7·332 | 7·911 | 8·596 | 9·268 | 4·091 | 6·013 | 7·854 | 9·613 | 11·290 | 12·885 | 14·399 | 3¼ |
| 7·636 | 8·241 | 8·957 | 9·660 | 4·254 | 6·259 | 8·181 | 10·022 | 11·781 | 13·458 | 15·053 | 3⅜ |
| 7·940 | 8·571 | 9·318 | 10·053 | 4·418 | 6·504 | 8·508 | 10·431 | 12·272 | 14·031 | 15·708 | 3½ |
| 8·243 | 8·901 | 9·679 | 10·446 | 4·581 | 6·750 | 8·836 | 10·840 | 12·763 | 14·604 | 16·362 | 3⅝ |
| 8·547 | 9·231 | 10·041 | 10·838 | 4·745 | 6·995 | 9·163 | 11·249 | 13·254 | 15·176 | 17·017 | 3¾ |
| 8·851 | 9·561 | 10·402 | 11·231 | 4·908 | 7·240 | 9·490 | 11·658 | 13·745 | 15·749 | 17·671 | 3⅞ |
| 9·154 | 9·891 | 10·763 | 11·624 | 5·072 | 7·486 | 9·817 | 12·067 | 14·235 | 16·322 | 18·326 | 4 |

## Table CLIII.—Standard List of Iron Welded Tubes as supplied by Tube Makers.

| External Diameter. | List Particulars. |  |  | Per Foot. |  | External Diameter. | List Particulars. |  |  | Per Foot. |  | External Diameter. | List Particulars. |  |  | Per Foot. |  |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
|  | Thickness. |  | Price per ft. | Surface. sq. ft. | Weight. lbs. |  | Thickness. |  | Price per ft. | Surface. sq. ft. | Weight. lbs. |  | Thickness. |  | Price per ft. | Surface. sq. ft. | Weight. lbs. |
|  | L.S.G. | Inch. |  |  |  |  | L.S.G. | Inch. |  |  |  |  | L.S.G. | Inch. |  |  |  |
|  |  |  | d. |  |  |  |  |  | d. |  |  |  |  |  | d. |  |  |
| 1 | 14 | ·080 | .. | ·262 | 0·771 | 3¾ | 10 | ·128 | 23·5 | ·982 | 4·855 | 8½ | 3 | ·252 | 151 | 2·225 | 21·77 |
| 1⅛ | 14 | ·080 | .. | ·295 | 0·875 | 4 | 9 | ·144 | 28·5 | 1·047 | 5·815 | 8¾ | 3 | ·252 | 156 | 2·291 | 22·43 |
| 1¼ | 13 | ·092 | 9·5 | ·327 | 1·116 | 4¼ | 9 | ·144 | 31·5 | 1·112 | 6·192 | 9 | 3 | ·252 | 160 | 2·357 | 23·09 |
| 1⅜ | 13 | ·092 | 9·5 | ·360 | 1·236 | 4½ | 9 | ·144 | 33·5 | 1·178 | 6·569 | 9¼ | 3 | ·252 | 175 | 2·423 | 23·75 |
| 1½ | 13 | ·092 | 9·5 | ·393 | 1·356 | 4¾ | 8 | ·160 | 40·5 | 1·243 | 7·691 | 9½ | 3 | ·252 | 182 | 2·490 | 24·41 |
| 1⅝ | 13 | ·092 | 9·75 | ·425 | 1·477 | 5 | 8 | ·160 | 43·5 | 1·309 | 8·109 | 9¾ | 3 | ·252 | 186 | 2·556 | 25·06 |
| 1¾ | 13 | ·092 | 10·0 | ·458 | 1·597 | 5¼ | 8 | ·160 | 47·0 | 1·376 | 8·528 | 10 | 3 | ·252 | 192 | 2·618 | 25·72 |
| 1⅞ | 13 | ·092 | 10·25 | ·491 | 1·718 | 5½ | 7 | ·176 | 54·0 | 1·442 | 9·813 | 10¼ | 2 | ·276 | 216 | 2·683 | 28·83 |
| 2 | 12 | ·104 | 11·75 | ·523 | 2·065 | 5¾ | 7 | ·176 | 59·0 | 1·507 | 10·27 | 10½ | 2 | ·276 | 222 | 2·749 | 29·55 |
| 2⅛ | 12 | ·104 | 12·25 | ·556 | 2·201 | 6 | 7 | ·176 | 62·0 | 1·572 | 10·73 | 10¾ | 1 | ·300 | 246 | 2·814 | 32·83 |
| 2¼ | 12 | ·104 | 12·25 | ·589 | 2·337 | 6¼ | 7 | ·176 | 70·0 | 1·638 | 11·20 | 11 | 1 | ·300 | 252 | 2·880 | 33·59 |
| 2⅜ | 11 | ·116 | 14·0 | ·622 | 2·744 | 6½ | 7 | ·176 | 73·0 | 1·703 | 11·66 | 11¼ | 1 | ·300 | 258 | 2·946 | 34·36 |
| 2½ | 11 | ·116 | 14·5 | ·654 | 2·896 | 6¾ | 7 | ·176 | 79·0 | 1·769 | 12·12 | 11½ | 1 | ·300 | 264 | 3·012 | 35·12 |
| 2⅝ | 11 | ·116 | 15·25 | ·687 | 3·048 | 7 | 7 | ·176 | 82·0 | 1·834 | 12·58 | 11¾ | 1 | ·300 | 270 | 3·078 | 35·88 |
| 2¾ | 11 | ·116 | 15·75 | ·720 | 3·200 | 7¼ | 5 | ·212 | 105 | 1·900 | 15·63 | 12 | 1 | ·300 | 276 | 3·142 | 36·65 |
| 2⅞ | 11 | ·116 | 16·50 | ·752 | 3·351 | 7½ | 5 | ·212 | 109 | 1·965 | 16·18 | 12½ | 1 | ·300 | 288 | 3·273 | 38·18 |
| 3 | 11 | ·116 | 17·0 | ·785 | 3·503 | 7¾ | 5 | ·212 | 113 | 2·030 | 16·74 | 13 | 1 | ·300 | 300 | 3·403 | 39·70 |
| 3¼ | 10 | ·128 | 20·0 | ·851 | 4·185 | 8 | 5 | ·212 | 117 | 2·093 | 17·29 | 13½ | 1 | ·300 | 312 | 3·534 | 41·23 |
| 3½ | 10 | ·128 | 22·0 | ·916 | 4·520 | 8¼ | 3 | ·252 | 147 | 2·159 | 21·11 | 14 | 1 | ·300 | 324 | 3·665 | 42·76 |

*Note.*—The above prices are for iron or steel lap-welded tubes, and are subject to a trade discount varying from 60 per cent. in good times to 75 in bad. The weight of steel tubes will be 2·38 per cent. or $\cdot/_{42}$ heavier than those given above. *Welded steel tubes* can be got at about the same rates as iron ones.

## Table CLIV.—Weight of Large Steel Tubes Solid Drawn or Welded, in lbs. per foot.

| External diameter. | Thickness in 64ths of an inch. | | | | | | | External diameter. | Thickness in 64ths of an inch. | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 8 | 9 | 10 | 11 | 12 | 14 | 16 | | 14 | 16 | 18 | 20 | 22 | 24 | 26 |
| ins. 10 | 13·19 | 14·81 | 16·43 | 18·05 | 19·65 | 22·86 | 26·04 | ins. 25 | 57·91 | 66·09 | 74·26 | 82·41 | 90·53 | 98·64 | 106·71 |
| 11 | 14·52 | 16·31 | 18·10 | 19·88 | 21·66 | 25·19 | 28·71 | 26 | 60·24 | 68·76 | 77·27 | 85·75 | 94·20 | 102·64 | 111·06 |
| 12 | 15·86 | 17·81 | 19·77 | 21·72 | 23·66 | 27·53 | 31·78 | 27 | 62·58 | 71·43 | 80·27 | 89·08 | 97·87 | 106·65 | 115·40 |
| 13 | 17·19 | 19·32 | 21·44 | 23·55 | 25·66 | 29·87 | 34·05 | 28 | 64·92 | 74·11 | 83·27 | 92·42 | 101·55 | 110·65 | 119·74 |
| 14 | 18·53 | 20·82 | 23·11 | 25·39 | 27·66 | 32·20 | 36·72 | 29 | ... | 76·78 | 86·28 | 95·76 | 105·22 | 114·66 | 124·08 |
| 15 | 19·86 | 22·32 | 24·77 | 27·22 | 29·67 | 34·54 | 39·39 | 30 | ... | 79·45 | 89·28 | 99·10 | 108·89 | 118·66 | 128·42 |
| 16 | 21·20 | 23·82 | 26·44 | 29·06 | 31·67 | 36·88 | 42·06 | 31 | ... | 82·12 | 92·29 | 102·44 | 112·56 | 122·67 | 132·76 |
| 17 | ... | 25·33 | 28·11 | 30·90 | 33·67 | 39·21 | 44·73 | 32 | ... | 84·79 | 95·29 | 105·77 | 116·23 | 126·67 | 137·10 |
| 18 | ... | 26·83 | 29·78 | 32·73 | 35·68 | 41·55 | 47·40 | 33 | ... | ... | 98·29 | 109·11 | 119·91 | 130·68 | 141·43 |
| 19 | ... | ... | 31·45 | 34·57 | 37·68 | 43·88 | 50·07 | 34 | ... | ... | 101·30 | 112·45 | 123·58 | 134·68 | 145·78 |
| 20 | ... | ... | 33·12 | 36·41 | 39·68 | 46·22 | 52·74 | 35 | ... | ... | 104·30 | 115·79 | 127·25 | 138·69 | 150·11 |
| 21 | ... | ... | ... | 38·24 | 41·68 | 48·56 | 55·41 | 36 | ... | ... | 107·31 | 119·13 | 130·92 | 142·70 | 154·45 |
| 22 | ... | ... | ... | 40·08 | 43·69 | 50·89 | 58·08 | 37 | ... | ... | 110·31 | 122·47 | 134·59 | 146·70 | 158·79 |
| 23 | ... | ... | ... | ... | 45·69 | 53·23 | 60·75 | 38 | ... | ... | 113·32 | 125·81 | 138·26 | 150·71 | 163·13 |
| 24 | ... | ... | ... | ... | 49·70 | 55·57 | 63·42 | 39 | ... | ... | 116·33 | 129·15 | 141·93 | 154·72 | 167·47 |

*N.B.*—Iron tubes will be ·/$_{42}$ lighter, that is $\frac{41}{42}$, or 0·976 of the above.

### Table CLV.—Weights of Seamless

| L. S. G. | | 4/0 | 3/0 | 2/0 | 0 | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|---|---|---|---|---|
| Inches | | ·400 | ·372 | ·348 | ·324 | ·300 | ·276 | ·252 | ·232 | ·212 |
| Millimetres | | 10·160 | 9·449 | 8·839 | 8·229 | 7·620 | 7·010 | 6·401 | 5·893 | 5·385 |
| Internal Diameter. | | Weight of a lineal | | | | | | | | |
| Inches | mm. | | | | | | | | | |
| ¾ | 19·0 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 1 | 25·4 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 1¼ | 31·7 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 1½ | 38·1 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 1¾ | 44·4 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 2 | 50·8 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 2¼ | 57·1 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 2½ | 63·5 | ... | ... | ... | ... | ... | ... | ... | ... | ... |
| 2¾ | 69·8 | ... | ... | ... | ... | ... | ... | ... | ... | 7·60 |
| 3 | 76·2 | ... | ... | ... | ... | ... | ... | ... | 9·07 | 8·24 |
| 3¼ | 82·5 | ... | ... | ... | ... | ... | ... | ... | 9·77 | 8·88 |
| 3½ | 88·9 | ... | ... | ... | ... | ... | ... | 11·44 | 10·47 | 9·52 |
| 3¾ | 95·2 | ... | ... | ... | ... | ... | 13·44 | 12·20 | 11·18 | 10·16 |
| 4 | 101·6 | ... | ... | ... | ... | ... | 14·28 | 12·96 | 11·88 | 10·80 |
| 4½ | 114·3 | ... | ... | ... | ... | 17·42 | 15·95 | 14·49 | 13·28 | 12·08 |
| 5 | 127·0 | ... | ... | ... | 20·87 | 19·23 | 17·62 | 16·01 | 14·68 | 13·37 |
| 5½ | 139·7 | ... | ... | ... | 22·83 | 21·05 | 19·29 | 17·54 | 16·09 | 14·65 |
| 6 | 152·4 | ... | ... | 26·72 | 24·79 | 22·86 | 20·95 | 19·06 | 17·49 | 15·93 |
| 6½ | 165·1 | ... | 30·93 | 28·83 | 26·75 | 24·68 | 22·62 | 20·58 | 18·89 | 17·21 |
| 7 | 177·8 | ... | 33·18 | 30·93 | 28·71 | 26·49 | 24·29 | 22·11 | 20·30 | 18·50 |
| 7½ | 190·5 | 38·23 | 35·43 | 33·04 | 30·67 | 28·31 | 25·96 | 23·63 | 21·70 | 19·78 |
| 8 | 203·2 | 40·65 | 37·68 | 35·14 | 32·63 | 30·12 | 27·63 | 25·16 | 23·10 | 21·06 |
| 8½ | 215·9 | 43·07 | 39·93 | 37·25 | 34·59 | 31·94 | 29·30 | 26·68 | 24·51 | 22·34 |
| 9 | 228·6 | 45·49 | 42·18 | 39·35 | 36·55 | 33·75 | 30·97 | 28·21 | 25·91 | 23·63 |
| 9½ | 241·3 | 47·91 | 44·43 | 41·46 | 38·51 | 35·57 | 32·64 | 29·73 | 27·31 | 24·91 |
| 10 | 254·0 | 50·33 | 46·68 | 43·56 | 40·47 | 37·38 | 34·31 | 31·25 | 28·72 | 26·19 |
| 10½ | 266·7 | 52·75 | 48·93 | 45·67 | 42·43 | 39·20 | 35·98 | 32·78 | 30·12 | 27·47 |
| 11 | 279·4 | 55·16 | 51·18 | 47·77 | 44·39 | 41·01 | 37·65 | 34·30 | 31·52 | 28·75 |
| 11½ | 292·1 | 57·58 | 53·43 | 49·88 | 46·35 | 42·83 | 39·32 | 35·83 | 32·93 | 30·04 |
| 12 | 304·8 | 60·00 | 55·68 | 51·98 | 48·31 | 44·64 | 40·99 | 37·35 | 34·33 | 31·32 |
| 12½ | 317·5 | 62·42 | 57·93 | 54·09 | 50·26 | 46·45 | 42·66 | 38·88 | 35·73 | 32·60 |
| 13 | 330·2 | 64·84 | 60·18 | 56·19 | 52·22 | 48·27 | 44·33 | 40·40 | 37·14 | 33·88 |
| 13½ | 342·9 | 67·26 | 62·43 | 58·30 | 54·18 | 50·08 | 46·00 | 41·92 | 38·54 | 35·17 |
| 14 | 355·6 | 69·68 | 64·68 | 60·40 | 56·14 | 51·90 | 47·67 | 43·45 | 39·94 | 36·45 |

**Mandril drawn Brazed Tubes weigh the**

**Ordinary Brazed Copper Tubes weigh rather more than Seamless Tubes, the**

## Copper Tubes.

| 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|
| ·192 | ·176 | ·160 | ·144 | ·128 | ·116 | ·104 | ·092 | ·080 | ·072 | ·064 |
| 4·877 | 4·470 | 4·064 | 3·658 | 3·251 | 2·946 | 2·642 | 2·377 | 2·032 | 1·829 | 1·626 |
| foot in pounds. | | | | | | | | | | |
| ... | ... | ... | ... | 1·36 | 1·21 | 1·07 | ·94 | ·80 | ·72 | ·63 |
| ... | ... | ... | ... | 1·75 | 1·57 | 1·39 | 1·21 | 1·04 | ·93 | ·82 |
| ... | ... | ... | 2·43 | 2·13 | 1·92 | 1·70 | 1·49 | 1·29 | 1·15 | 1·02 |
| ... | ... | 3·21 | 2·86 | 2·52 | 2·27 | 2·02 | 1·77 | 1·53 | 1·37 | 1·21 |
| ... | ... | 3·70 | 3·30 | 2·91 | 2·62 | 2·33 | 2·05 | 1·77 | 1·59 | 1·40 |
| ... | 4·63 | 4·18 | 3·73 | 3·29 | 2·97 | 2·65 | 2·33 | 2·01 | 1·80 | 1·60 |
| 5·67 | 5·16 | 4·66 | 4·17 | 3·68 | 3·32 | 2·96 | 2·61 | 2·25 | 2·02 | 1·79 |
| 6·25 | 5·70 | 5·15 | 4·61 | 4·07 | 3·67 | 3·28 | 2·88 | 2·50 | 2·24 | 1·98 |
| 6·83 | 6·23 | 5·63 | 5·04 | 4·46 | 4·02 | 3·59 | 3·16 | 2·74 | 2·46 | 2·18 |
| 7·41 | 6·76 | 6·12 | 5·48 | 4·84 | 4·37 | 3·90 | 3·44 | 2·98 | 2·68 | 2·37 |
| 7·99 | 7·29 | 6·60 | 5·91 | 5·23 | 4·72 | 4·22 | 3·72 | 3·22 | 2·89 | 2·57 |
| 8·58 | 7·83 | 7·08 | 6·35 | 5·62 | 5·07 | 4·53 | 4·00 | 3·46 | 3·11 | 2·76 |
| 9·16 | 8·36 | 7·57 | 6·78 | 6·00 | 5·42 | 4·85 | 4·28 | 3·71 | 3·33 | 2·95 |
| 9·74 | 8·89 | 8·05 | 7·22 | 6·39 | 5·78 | 5·16 | 4·55 | 3·95 | 3·55 | 3·15 |
| 10·90 | 9·96 | 9·02 | 8·09 | 7·17 | 6·48 | 5·79 | 5·11 | 4·43 | 3·98 | 3·53 |
| 12·06 | 11·02 | 9·99 | 8·96 | 7·94 | 7·18 | 6·42 | 5·67 | 4·92 | 4·42 | 3·92 |
| 13·22 | 12·08 | 10·96 | 9·83 | 8·71 | 7·88 | 7·05 | 6·22 | 5·40 | 4·85 | 4·31 |
| 14·38 | 13·15 | 11·92 | 10·70 | 9·49 | 8·58 | 7·68 | 6·78 | 5·88 | 5·29 | 4·69 |
| 15·54 | 14·21 | 12·89 | 11·57 | 10·26 | 9·28 | 8·31 | 7·34 | 6·37 | 5·72 | 5·08 |
| 16·70 | 15·28 | 13·86 | 12·44 | 11·04 | 9·99 | 8·94 | 7·89 | 6·85 | 6·16 | 5·47 |
| 17·87 | 16·34 | 14·83 | 13·32 | 11·81 | 10·69 | 9·57 | 8·45 | 7·34 | 6·60 | 5·86 |
| 19·03 | 17·41 | 15·79 | 14·19 | 12·59 | 11·39 | 10·20 | 9·01 | 7·82 | 7·03 | 6·24 |
| 20·19 | 18·47 | 16·76 | 15·06 | 13·36 | 12·09 | 10·82 | 9·56 | 8·30 | 7·47 | 6·63 |
| 21·35 | 19·54 | 17·73 | 15·93 | 14·13 | 12·79 | 11·45 | 10·12 | 8·79 | 7·90 | 7·02 |
| 22·51 | 20·60 | 18·70 | 16·80 | 14·91 | 13·49 | 12·08 | 10·68 | 9·27 | 8·34 | ... |
| 23·67 | 21·67 | 19·67 | 17·67 | 15·68 | 14·20 | 12·71 | 11·23 | 9·76 | 8·77 | ... |
| 24·83 | 22·73 | 20·63 | 18·54 | 16·46 | 14·90 | 13·34 | 11·79 | 10·24 | ... | ... |
| 26·00 | 23·80 | 21·60 | 19·41 | 17·23 | 15·60 | 13·97 | 12·34 | 10·72 | ... | ... |
| 27·16 | 24·86 | 22·57 | 20·28 | 18·01 | 16·30 | 14·60 | 12·90 | ... | ... | ... |
| 28·32 | 25·92 | 23·54 | 21·16 | 18·78 | 17·00 | 15·23 | 13·46 | ... | ... | ... |
| 29·48 | 26·99 | 24·50 | 22·03 | 19·55 | 17·70 | 15·86 | ... | ... | ... | ... |
| 30·64 | 28·05 | 25·47 | 22·90 | 20·33 | 18·41 | 16·49 | ... | ... | ... | ... |
| 31·80 | 29·12 | 26·44 | 23·77 | 21·10 | 19·11 | ... | ... | ... | ... | ... |
| 32·96 | 30·18 | 27·41 | 24·64 | 21·88 | 19·81 | ... | ... | ... | ... | ... |

**same as Seamless Tubes.**
**difference varying with the thickness, diameter, and class of joint.**

## Table CLVI.—Whitworth Gas Threads.

| Diameter of pipe (inside). | Diameter over threads. | Diameter at bottom of thread. | Number of threads per inch. |
|---|---|---|---|
| 1/8" | ·3825 | ·3367 | 28 |
| 1/4 | ·5180 | ·4506 | 19 |
| 3/8 | ·6563 | ·5889 | 19 |
| 1/2 | ·8257 | ·7342 | 14 |
| 5/8 | ·9022 | ·8107 | 14 |
| 3/4 | 1·041 | ·9495 | 14 |
| 7/8 | 1·189 | 1·0975 | 14 |
| 1 | 1·309 | 1·1925 | 11 |
| 1 1/8 | 1·492 | 1·3755 | 11 |
| 1 1/4 | 1·65 | 1·5335 | 11 |
| 1 3/8 | 1·745 | 1·6285 | 11 |
| 1 1/2 | 1·8825 | 1·766 | 11 |
| 1 5/8 | 2·021 | 1·9045 | 11 |
| 1 3/4 | 2·047 | 1·9305 | 11 |
| 1 7/8 | 2·245 | 2·1285 | 11 |
| 2 | 2·347 | 2·2305 | 11 |
| 2 1/4 | 2·5875 | 2·471 | 11 |
| 2 1/2 | 3·0013 | 2·8848 | 11 |
| 2 3/4 | 3·247 | 3·1305 | 11 |
| 3 | 3·485 | 3·3685 | 11 |
| 3 1/4 | 3·6985 | 3·582 | 11 |
| 3 1/2 | 3·912 | 3·7955 | 11 |
| 3 3/4 | 4·1225 | 4·009 | 11 |
| 4 | 4·339 | 4·2225 | 11 |

## Table CLVII.—Weight of Brass Condenser Tubes.

| L.S.G. | | 14 | 15 | 16 | 17 | 18 | 19 | 20 |
|---|---|---|---|---|---|---|---|---|
| Inches. | | ·080 | ·072 | ·064 | ·056 | ·048 | ·040 | ·036 |
| Millimetres. | | 2·032 | 1·829 | 1·626 | 1·422 | 1·219 | 1·016 | ·914 |
| Ext. Diameter. Ins. | Ext. Diameter. mm. | Weight of a Lineal Foot in pounds. | | | | | | |
| 5/8 | 15·9 | ·51 | ·46 | ·42 | ·37 | ·32 | ·27 | ·25 |
| 3/4 | 19·0 | ·62 | ·57 | ·51 | ·45 | ·39 | ·33 | ·30 |
| 7/8 | 22·2 | 74 | ·67 | ·60 | ·53 | ·46 | ·39 | ·35 |
| 1 | 25·4 | ·86 | ·78 | ·70 | ·62 | ·53 | ·45 | ·40 |

*Note.*—**The above weights are for tubes containing 70 per cent. copper.**

## Table CLVIII.—Weight of Lead Pipes.

| Thickness in inches. | Weight in pounds per foot run. | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 2″ | 2¼″ | 2½″ | 2¾″ | 3″ | 3¼″ | 3½″ | 3¾″ | 4″ | 4½″ | 5″ |
| ¼ | 8·7 | 9·7 | 10·6 | 11·6 | 12·6 | 13·5 | 14·5 | 15·5 | 16·4 | 18·4 | 20·3 |
| 5/16 | ... | ... | 13·6 | 14·8 | 16·0 | 17·2 | 18·4 | 19·7 | 20·9 | 23·3 | 25·7 |
| ⅜ | ... | ... | ... | ... | ... | 21·0 | 22·5 | 23·9 | 25·4 | 28·3 | 31·2 |
| 7/16 | ... | ... | ... | ... | ... | ... | ... | ... | ... | 33·4 | 36·8 |

Lead pipes of these sizes and weights are usually manufactured in 10 ft. lengths.

## Table CLIX.—Weight of Sheet Metals.

| Thickness L.S.G. | Weight in pounds per square foot. | | | | | |
|---|---|---|---|---|---|---|
| | Steel. | Iron. | Copper. | Brass. | Lead. | Zinc. |
| 7/0 | 20·40 | 20·00 | 22·83 | 21·98 | 29·65 | 18·72 |
| 6/0 | 18·93 | 18·56 | 21·19 | 20·39 | 27·51 | 17·38 |
| 5/0 | 17·63 | 17·28 | 19·73 | 18·99 | 25·62 | 16·18 |
| 4/0 | 16·32 | 16·00 | 18·27 | 17·58 | 23·72 | 14·98 |
| 3/0 | 15·18 | 14·88 | 16·99 | 16·35 | 22·06 | 13·93 |
| 2/0 | 14·20 | 13·92 | 15·89 | 15·30 | 20·64 | 13·03 |
| 0 | 13·22 | 12·96 | 14·80 | 14·24 | 19·21 | 12·13 |
| 1 | 12·24 | 12·00 | 13·70 | 13·19 | 17·79 | 11·23 |
| 2 | 11·26 | 11·04 | 12·60 | 12·13 | 16·37 | 10·34 |
| 3 | 10·28 | 10·08 | 11·51 | 11·08 | 14·94 | 9·44 |
| 4 | 9·47 | 9·28 | 10·59 | 10·20 | 13·76 | 8·69 |
| 5 | 8·65 | 8·48 | 9·68 | 9·32 | 12·57 | 7·94 |
| 6 | 7·83 | 7·68 | 8·77 | 8·44 | 11·38 | 7·19 |
| 7 | 7·18 | 7·04 | 8·04 | 7·74 | 10·44 | 6·59 |
| 8 | 6·53 | 6·40 | 7·31 | 7·03 | 9·49 | 5·99 |
| 9 | 5·88 | 5·76 | 6·58 | 6·33 | 8·54 | 5·39 |
| 10 | 5·22 | 5·12 | 5·84 | 5·63 | 7·59 | 4·79 |
| 11 | 4·73 | 4·64 | 5·30 | 5·10 | 6·88 | 4·34 |
| 12 | 4·24 | 4·16 | 4·75 | 4·57 | 6·17 | 3·89 |
| 13 | 3·75 | 3·68 | 4·20 | 4·04 | 5·45 | 3·44 |
| 14 | 3·26 | 3·20 | 3·65 | 3·52 | 4·74 | 3·00 |
| 15 | 2·94 | 2·88 | 3·29 | 3·16 | 4·27 | 2·70 |
| 16 | 2·61 | 2·56 | 2·92 | 2·81 | 3·79 | 2·40 |
| 17 | 2·28 | 2·24 | 2·56 | 2·46 | 3·32 | 2·10 |
| 18 | 1·96 | 1·92 | 2·19 | 2·11 | 2·85 | 1·80 |
| 19 | 1·63 | 1·60 | 1·83 | 1·76 | 2·37 | 1·50 |
| 20 | 1·47 | 1·44 | 1·64 | 1·58 | 2·13 | 1·35 |
| 21 | 1·31 | 1·28 | 1·46 | 1·41 | 1·90 | 1·20 |
| 22 | 1·14 | 1·12 | 1·28 | 1·23 | 1·66 | 1·05 |

## Table CLX.—Weights of Engines and Boilers, &c.

| Description of Ship and Machinery. | Weight in Pounds per H.P. | | | I.H.P. Per Ton Total Weight. |
|---|---|---|---|---|
| | Engines, Propellers, &c. | Boilers, &c. | Total Machinery. | |
| (1) Battleships and large cruisers, triple engines, tank boilers, 155 lbs. pressure | 115 | 150 | 265 | 8·45 |
| (2) Battleships and large cruisers, triple engines, water tube boilers, 250 lbs. pressure | 115 | 131 | 246 | 9·11 |
| (3) Battleships and large cruisers, turbines, mixed boilers, 215 lbs. pressure | .. | .. | 177 | 12·68 |
| (4) Large high-speed cruisers, triples, tank boilers, 155 lbs., M.F.D. | .. | .. | 202 | 11·09 |
| (5) Large high-speed cruisers, triple, water tube boilers, 250 lbs., M.F.D. | .. | .. | 188 | 11·92 |
| (6) Large high-speed cruisers, triples, mixed boilers, 210 lbs., M.F.D. | .. | .. | 215 | 10·42 |
| (7) Large high-speed cruisers, turbines, water tube boilers, 210 lbs., M.F.D. | .. | .. | .. | .. |
| (8) Moderate size cruisers with triple engines, tank boilers, 155 lbs., M.F.D. | .. | .. | 195 | 11·5 |
| (9) Moderate size cruisers, with triple engines, water tube boilers, 250 lbs., M.F.D. | .. | .. | 185 | 12·1 |
| (10) Moderate size cruisers with turbines, water tube boilers, 210 lbs., M.F.D. | .. | .. | 91·8 | 24·2 |
| (11) Scouts, &c., triple engines, water tube boilers, 300 lbs. | .. | .. | 113 | 19·8 |
| (12) Scouts, &c., turbines, water tube boilers | .. | .. | 59·7 | 37·5 |
| (13) Destroyers, triple engines, water tube boilers | 24·0 | 24·0 | 48·0 | 46·7 |
| (14) Destroyers, turbines, water tube boilers | 16·4 | 21·6 | 37·8 | 59·3 |
| (15) Largest and fastest express Atlantic steamers, reciprocators, tank boilers | .. | .. | 345 | 6·49 |
| (16) Largest and fastest express Atlantic steamers, turbines, tank boilers | .. | .. | .. | .. |
| (17) Large expresses: 20 knots, passenger and cargo, Atlantic, reciprocators | .. | .. | 440 | 5·09 |
| (18) Large expresses: 20 knots, passenger and cargo, Atlantic, turbines | 202 | 223 | 425 | 5·25 |
| (19) Express channel and excursion steamers, reciprocators | .. | .. | 213 | 10·52 |
| (20) Express channel and excursion steamers, turbines | 47 | 121 | 168 | 13·33 |
| (21) Large cargo and passenger steamers, Eastern service, reciprocators | .. | .. | 330 | 6·79 |
| (22) Ordinary cargo steamers, triples and quadruples | .. | .. | 462 | 4·85 |
| (23) Large express short-service paddle steamers, compound diagonal engines | .. | .. | 285 | 7·86 |
| (24) Ordinary river service, moderate size, compound diagonal engines | 185 | 180 | 365 | 6·14 |

The machinery of torpedo boats with loco. boilers weighed about 30 to 35 lbs. per I.H.P. in the engine-room, and about 45 to 50 lbs. in the boiler-room, or say 81 lbs. per I.H.P. for both engine- and boiler-room. When water-tube boilers are used, the boiler-room weights are reduced to about 33 lbs. per I.H.P. As the power increases the weight per I.H.P. slightly decreases, so that in somewhat similar twin-screw vessels with water-tube boilers indicating 6000 to 8000 horse-power, 48 to 50 lbs. per I.H.P. covers everything. The machinery of some of the 30-knot destroyers was only 43 lbs. per I.H.P. with reciprocating engines and high forced draughts.

The machinery of Naval pinnaces weighs from 90 to 115 lbs., or say ·045 ton per I.H.P., and that of Naval cutters from 150 to 190 lbs., or say ·076 ton per I.H.P.,—the smaller sizes weighing rather heavier than the larger. Since the adoption of water-tube boilers in pinnaces, these weights (which refer to boats with tank-type boilers) have been reduced.

The weights of paddle engines depend to some extent on the service for which the vessel is intended, since for cross-channel work the wheels will be much heavier than for comparatively smooth water work.

## CENTRE OF GRAVITY OF MACHINERY, &c.

When the common centre of gravity of a number of detached bodies,—such as the various portions of the machinery of a steamship,—is required, it is obtained as follows:—

1. **Longitudinal Position of Centre of Gravity.**—On the longitudinal section drawing of the machinery, near where it is supposed the longitudinal position of the centre of gravity will be, draw a vertical line, to represent the edge of a plane cutting the ship transversely. Then, take the weight of each portion of the machinery, and multiply it by the distance of its centre of gravity from this transverse plane; distinguish the resulting quantities, or moments, by the signs + and −, according as they happen to be derived from the right or left side of the plane; and add together all the plus quantities, and all the minus quantities. Then obtain the resultant, by subtracting the less quantity from the greater, and place before it the sign of the greater.

If this resultant moment be now divided by the *total* weight of the whole system of bodies, on both sides of the plane, the quotient will be the distance of the longitudinal position of the centre of gravity from the plane; and it will lie to the right, or to the left, of the plane, according as the sign of the quotient is plus or minus.

2. **Transverse Position of Centre of Gravity.**—Assume another plane, perpendicular to the first, and represent it by a line on the transverse section drawings, and then proceed exactly as described above.

3. **Vertical Position of Centre of Gravity.**—Assume a third plane, perpendicular to both those previously used (parallel to the water-line, that is), represent it by a line or lines as before, and again calculate the position as described above.

# MISCELLANEOUS TABLES.

## Table CLXI.—Surface of Tubes in square feet per foot run.

| Diamr. in inches. | 0 | ⅛ | ¼ | ⅜ | ½ | ⅝ | ¾ | ⅞ |
|---|---|---|---|---|---|---|---|---|
| 0 | ... | ·0327 | ·0654 | ·0982 | ·1309 | ·1636 | ·1963 | ·2291 |
| 1 | ·2618 | ·2945 | ·3272 | ·3600 | ·3927 | ·4254 | ·4581 | ·4909 |
| 2 | ·5236 | ·5563 | ·5890 | ·6218 | ·6545 | ·6872 | ·7200 | ·7527 |
| 3 | ·7854 | ·8181 | ·8508 | ·8836 | ·9163 | ·9490 | ·9817 | 1·0145 |
| 4 | 1·0472 | 1·0799 | 1·1126 | 1·1454 | 1·1781 | 1·2108 | 1·2435 | 1·2763 |
| 5 | 1·3090 | 1·3417 | 1·3744 | 1·4072 | 1·4399 | 1·4726 | 1·5053 | 1·5381 |

## Bulk and Weight, &c., of Water.

## Table CLXII.—Fresh Water.

| Cubic inches. | Cubic feet. | Pounds. | Gallons. | Ton. |
|---|---|---|---|---|
| 1 | ... | ·036 | ... | ... |
| 27·7 | ·016 | 1·0 | ·1 | ... |
| 277 | ·160 | 10·0 | 1·0 | ·00446 |
| 1728 | 1·0 | 62·4 | 6·24 | ·0279 |
| ... | 35·9 | 2240 | 224 | 1·0 |

The above figures are very nearly correct for 62° F. : for bulk of water at other temperatures see page 400.

## Table CLXIII.—Salt Water.

| Cubic inches. | Cubic feet. | Pounds. | Gallons. | Ton. |
|---|---|---|---|---|
| 1 | ... | ·037 | ... | ... |
| 27 | ·0156 | 1·0 | ·097 | ... |
| 277 | ·160 | 10·26 | 1·0 | ·00458 |
| 1728 | 1·0 | 64·0 | 6·24 | ·0285 |
| ... | 35·0 | 2240 | 218·3 | 1·0 |

The above figures are very nearly correct for 62° F. : for bulk of water at other temperatures see page 400.

## Table CLXIV.—Solid Matter deposited from Feed Water—

*Prof. V. Lewes.*

| Constituents. | River. | Tidal Rivers. | Sea. |
|---|---|---|---|
| Calcic carbonate, . . . . | 75·85 | 43·65 | 0·97 |
| ,, sulphate, . . . . | 3·68 | 34·78 | 85·53 |
| Magnesic hydrate, . . . . | 2·56 | 4·34 | 3·39 |
| Sodic chloride, . . . . | 0·45 | 0·56 | 2·79 |
| Silica, . . . . . . | 7·66 | 7·52 | 1·10 |
| Oxides of iron and alumina, . | 2·96 | 3·44 | 0·32 |
| Organic matter, . . . . | 3·64 | 1·55 | trace |
| Moisture, . . . . . | 3·20 | 4·16 | 5·90 |
| | 100·00 | 100·00 | 100·00 |

## Table CLXV.—Sea-Water, Solid Contents of.

Sea water contains certain mineral salts in solution amounting to 3·4 per cent. of its weight on the average, and is at the various parts of the globe as follows :—

| | | | |
|---|---|---|---|
| Arctic Ocean . | 2·83 per cent. | Red Sea . . | 4·30 per cent. |
| North Atlantic . | 4·26 ,, | Dead Sea . . | 38·66 ,, |
| South ,, . | 4·12 ,, | English Channel | 3·55 ,, |
| Equator . . | 3·94 ,, | Irish ,, | 3·38 ,, |
| Mediterranean . | 3·94 ,, | Upper Baltic . | 0·66 ,, |
| Sea of Marmora . | 4·20 ,, | Cronstadt . . | 2·00 ,, |
| Black Sea . . | 2·16 ,, | | |

## Table CLXVI.—Composition of Solids in Solution in Sea-Water.

| Salts in Sea Water. | Kingston Harbour. Main and Brown. | Average. D. K. Clerk. | English Channel. Thomson. |
|---|---|---|---|
| Chloride of sodium, . . | 71·32 | 73·60 | 76·70 |
| Chloride of potassium, . | — | — | 2·17 |
| Chloride of magnesium, . | 10·79 | 9·80 | 10·40 |
| Sulphate of magnesium, . | 5·30 | 15·73 | 6·56 |
| Sulphate of lime, . . | 4·87 | 0·29 | 4·0 |
| Carbonate of lime, . . | 1·73 | 0·58 | 0·09 |
| Bromide of magnesium, . | 0·60 | — | 0·08 |
| Organic matter, . . | 5·27 | — | — |

## Table CLXVII.—Weight of Sea-Water at various Ports and Harbours of the United Kingdom, taken at high-water and given in ounces per cubic foot; fresh-water being 1000.

| Locality. | Weight | Locality. | Weight |
|---|---|---|---|
| Thames at Limehouse | 1000 | Plymouth Sound | 1025 |
| Do. Victoria Dock | 1005 | Weymouth | 1024 |
| Do. Gravesend | 1019 | Southampton Extension Dock | 1023 |
| Humber at Goole Dock | 1000 | Portsmouth, Camber | 1024 |
| Do. off Hull | 1016 | Dublin | 1024 |
| Do. off Grimsby | 1020 | Kingstown | 1025 |
| Tyne at Low Walker | 1013 | Queenstown Harbour | 1025 |
| Do. Tyne Dock | 1023 | Cork | 1000 |
| Tees at Stockton | 1005 | Youghal | 1025 |
| Do. Middlesbrough | 1012 | Limerick | 1000 |
| Do. Hartlepool | 1025 | Belfast Harbour | 1011 |
| Sunderland Harbour | 1024 | Glasgow, Bowling | 1000 |
| Mersey at Connahs Quay | 1018 | Greenock, Tail of Bank | 1022 |
| Do. Liverpool | 1021 | Port Glasgow, Dock Gates | 1013 |
| Milford Haven | 1023 | Troon | 1023 |
| Avonmouth Dock, Bristol | 1011 | Campbeltown | 1024 |
| Portishead Dock, do. | 1015 | Fraserburgh Harbour | 1025 |
| Cardiff, Roath Basin | 1008 | Peterhead Harbour | 1024 |
| Do. Penarth Basin | 1018 | Dundee | 1021 |
| Barrow, Buccleugh Dock | 1017 | St Andrews | 1025 |
| Whitehaven, Queen's Dock | 1024 | Leith and Granton, Forth at | 1025 |
| Birkenhead Float | 1020 | Bo'ness Dock | 1021 |

## Table CLXVIII.—Boiling Points, &c., of Sea-Water.

| Proportion of salt in the water. | Temperature, Fahrenheit, at which it boils. | Specific Gravity. |
|---|---|---|
| $^{1}/_{32}$ | 213·2° | 1·029 |
| $^{2}/_{32}$ | 214·4° | 1·058 |
| $^{3}/_{32}$ | 215·5° | 1·087 |
| $^{4}/_{32}$ | 216·7° | 1·116 |
| $^{5}/_{32}$ | 217·9° | 1·145 |
| $^{6}/_{32}$ | 219·1° | 1·174 |
| $^{7}/_{32}$ | 220·3° | 1·203 |
| $^{8}/_{32}$ | 221·5° | 1·232 |
| $^{9}/_{32}$ | 222·7° | 1·261 |
| $^{10}/_{32}$ | 223·8° | 1·290 |
| $^{11}/_{32}$ | 225·0° | 1·319 |
| $^{12}/_{32}$ | 226·1° | 1·348 |

*Note.*—Freezing Mixtures are made as follows :—(*a*) Mix one part of common salt with two parts of pounded ice ; the temperature will be −4° F. or −20° C.; (*b*) mix three parts of chloride of calcium with two parts of snow ; the temperature then is −50° F. or −45·5° C.

## Table CLXIX.—Viscosity of Oils at Various Temperatures (Water at 68° F. being 0·01028)—*Archbutt and Deeley.*

| Name of Oil. | 60° F. | 100° F. | 150° F. | 212° F. |
|---|---|---|---|---|
| Sperm oil, | 0·420 | 0·185 | 0·085 | 0·046 |
| Olive ,, | 1·008 | 0·377 | 0·154 | 0·070 |
| Rape or colza oil, | 1·118 | 0·422 | 0·177 | 0·080 |
| Castor oil, | ... | 2·729 | 0·605 | 0·169 |
| Scotch mineral oil "865," | 0·146 | 0·066 | 0·036 | ... |
| ,, ,, "890," | 0·509 | 0·183 | 0·069 | ... |
| Russian light machinery oil, | 1·156 | 0·307 | 0·099 | 0·043 |
| ,, heavy ,, | 3·592 | 0·762 | 0·196 | 0·066 |
| American spindle oil, | 0·727 | 0·236 | 0·086 | 0·039 |
| ,, pale ,, "903/7," | 1·138 | 0·342 | 0·115 | 0·049 |
| ,, ,, ,, "907/12," | 1·479 | 0·419 | ... | ... |
| ,, red engine oil "910/20," | 1·915 | 0·496 | 0·150 | 0·058 |
| ,, "Bayonne" engine oil "912/15," | 2·172 | 0·572 | 0·173 | 0·063 |
| ,, dark medium machinery oil, | 3·046 | 0·705 | 0·210 | 0·076 |
| "Valvoline," | ... | 2·406 | 0·605 | 0·187 |
| American filtered cylinder oil "888," | ... | 3·702 | ... | 0·238 |
| ,, dark ,, ,, "900," | ... | 6·264 | ... | 0·314 |

## Table CLXX.

The oils generally used in connection with ordinary machinery will give satisfactory results if they comply with the following specifications:—

| Items. | Specific Gravity at 60° F. | Viscosity at 140° F. Secs. | Viscosity at 200° F. Secs. | Viscosity at 300° F. Secs. | Flashing Point. | Loss at 212° F. in 2 hours not more than | Saponifiable Oil % | Free Acid not exceeding |
|---|---|---|---|---|---|---|---|---|
| *Machinery oil (ordinary) | Abt. 0·90 | 165–180 | 60–65 | — | Not below 370° F. | 0·15 % | 10–12 % | 0·2 % |
| Machinery oil for forced lubrication | Abt. 0·91 | Abt. 85 | Abt. 44·5 | Abt. 31 | 390°–400° F. | 0·2 % | Nil. | 0·1 % |
| †Cylinder oil . . . | Abt. 0·90 | 180–200 | 65–70 | 35–40 | Not under 395° F. | 0·1 % | 5–6 % | 0·1 % |
| *Gas engine oil . . . | Abt. 0·91 | 82–87 | 44–48 | 32–34 | ,, | 0·2 % | 5–6 % | 0·1 % |
| For heavy or troublesome bearings either pure castor oil or oil to the following specification may be used. | Abt. 0·91 | 180–200 | 65–70 | — | Not below 350° F. | 0·2 % | 18–22 % | 0·4 % |

### *NOTES.*

The gas engine oil will also serve very well for light high-speed machinery.

Viscosities to be taken with Boverton Redwood's apparatus.

Flash points to be determined by means of the Pensky-Martens apparatus.

The hydrocarbon oils should be free from grit or dirt, acid, alkali, resin oil, bituminous matter, and all artificial thickening agents; they should also form a clear mixture with rape oil at ordinary temperatures. The saponifiable oil should be lard or tallow oil (not any seed oil), and should not contain more than 2 per cent. of free fatty acid.

The hydrocarbon oils, after being kept at 25° F. for one hour in a ¾-inch. int. dia. test tube, should flow when the tube is inclined.

* These oils should be either distilled or charcoal refined or a blend of these types with the addition of the necessary percentage of pure lard or tallow oil only (no seed oil should be accepted), and should be practically transparent.

† A proportion of "natural" oil may be used in producing the necessary blends, but the oil supplied must be free from bituminous matter.

As viscosity at 300 F. is troublesome to ascertain, the oil may, as an alternative test, be kept at 365° F. for 24 hours and should not suffer any appreciable change in properties, though it may be slightly darkened in colour.

## Table CLXXI.—Boiling, Setting, and Flash Points.

| Name or Brand of Oil. | Sets at | Flashes at | Boils at |
|---|---|---|---|
| | Degrees F. | Degrees F. | Degrees F. |
| American cylinder, sp. gr. ·902, . . | 25 | 585 | 650 |
| ,, ,, ,, ·893, . . | 25 | 424 | 600 |
| Valvoline ,, . . . . . . | ... | ... | 700 |
| American machine, sp. gr. ·897, . . | 25 | 402 | 600 |
| Russian ,, ,, ·909, . . | 0 to 10 | 380 | ... |
| Scotch shale oil, . . . . . . | 32 | ... | ... |
| Cotton-seed oil, . . . . . . | 34 to 50 | ... | 580 |
| Castor oil, . . . . . . . | 14 | ... | 556 |
| Lard oil, . . . . . . . | 25 to 42 | ... | ... |
| Linseed oil, . . . . . . . | 5 ,, 17 | ... | 574 |
| Neat's foot oil, . . . . . . . | 32 ,, 50 | ... | ... |
| Olive oil, . . . . . . . | 21 ,, 39 | ... | ... |
| Rape or colza oil, . . . . . | 10 ,, 28 | ... | 625 |
| Sperm oil, . . . . . . . | 32 | 450 | 600 |
| Palm oil, . . . . . . . . | 81 to 106 | ... | ... |
| Whale oil, . . . . . . | 32 | 440 | 620 |
| Vaseline oil, . . . . . | 104 to 120 | 640 | ... |
| Tallow (Russian), . . . . . | 107 ,, 119 | 540 | 660 |

## Table CLXXII.—Co-efficients of Friction.

| Nature of the Surfaces. | Quite Dry. | Greasy Dry. | Well Lubricated with | | | Constant Flow of Oil. |
|---|---|---|---|---|---|---|
| | | | Oil. | Tallow or Soap. | Water. | |
| Wood on wood (dry), . . . | 0·25 to 0·5 | ... | ... | 0·2 to 0·04 | ... | ... |
| Metal on hard wood, . . | ... | 0·15 | 0·07 to 0·08 | ... | ... | 0·05 |
| Metals on oak, . . . . (Rk) | 0.5 to 0·6 | 0·24 to 0·26 | ... | 0·2 | ... | ... |
| ,, metals only, . . ,, | ... | 0·15 to 0·2 | ... | 0·14 | 0·3 | ... |
| ,, ,, smooth, ,, | ... | 0·07 to 0·08 | 0·05 | ... | ... | 0·03 |
| Bronze on lignum vitæ, . . ,, | ... | ... | ... | ... | 0·05 | ... |
| Iron ,, ,, . . . | .. | ... | ... | ... | 0·05 | ... |
| Bronze on bronze sliding, . (S) | ... | ... | ... | ... | 0·38 | ... |
| ,, ,, ,, . . (Rn) | ... | 0·175 | ... | ... | ... | ... |
| Steel ,, ,, . . . ,, | ... | 0·139 | ... | ... | ... | ... |
| Wrought iron on bronze sliding, ,, | ... | 0·135 | ... | ... | ... | ... |
| Cast ,, ,, ,, ,, | ... | 0·141 | ... | ... | ... | ... |
| Steel on cast iron, . . . ,, | ... | 0·151 | ... | ... | ... | ... |

## CLXXII.—Co-efficients of Friction—*continued.*

| Nature of the Surfaces. | Quite Dry. | Greasy Dry. | Well Lubricated with |  |  | Constant Flow of Oil. |
|---|---|---|---|---|---|---|
|  |  |  | Oil. | Tallow or Soap. | Water. |  |
| Steel on wrought iron, . . (Rn) | ... | 0·189 | ... | ... | ... | ... |
| Wrought iron on cast iron, . ,, | ... | 0·170 | ... | ... | ... | ... |
| ,, ,, tin, . . ,, | ... | 0·181 | ... | ... | ... | ... |
| ,, ,, magnolia, (RHS) | ... | 0·060 | ... | ... | ... | ... |
| ,, ,, white metal, . | ... | .. | 0·025 | ... | ... | 0·01 to 0·015 |
| Loco axles, white metal, . (W) | ... | ... | 0·017 | ... | ... | ... |
| Shafts on bronze, . . . (Rn) | ... | ... | ... | 0·028 | ... | ... |
| Leather on metals, . . . (Rk) | 0·56 | 0·23 | 0·15 | ... | 0·36 | ... |
| Wrought-iron shaft on cast-iron bearing, speed 10 feet per minute, | .. | ... | 0·079 | ... | ... | ... |
| Wrought-iron shaft on cast-iron bearing, speed 60 feet per minute, | ... | ... | 0·052 | ... | ... | ... |
| Wrought-iron shaft on cast-iron bearing, speed 100 feet per minute, | ... | ... | 0·050 | ... | ... | .. |
| Load 50 lbs. p. sq. in., speed 50 ft., (G) | ... | ... | ... | ... | ... | ·0032 |
| ,, 50 ,, ,, ,, 110 ,, ,, | ... | ... | ... | .. | ... | ·0064 |
| ,, 50 ,, ,, ,, 190 ,, ,, | ... | ... | ... | ... | ... | ·0106 |
| ,, 150 ,, ,, ,, 50 ,, ,, | ... | .. | ... | ... | ... | ·0017 |
| ,, 150 ,, ,, ,, 110 ,, ,, | ... | ... | ... | .. | ... | ·0024 |
| ,, 150 ,, ,, ,, 190 ,, ,, | ... | .. | ... | ... | ... | ·0047 |
| Speed of 10 ft. p. min., load 50 lbs., ,, | ... | ... | ... | ... | ... | ·0009 |
| ,, 10 ,, ,, 75 ,, ,, | ... | ... | ... | ... | ... | ·0007 |
| ,, 10 ,, ,, 150 ,, ,, | .. | ... | ... | ... | ... | ·0250 |
| ,, 15 ,, ,, 50 ,, ,, | ... | ... | .. | ... | ... | ·0012 |
| ,, 15 ,, ,, 150 ,, ,, | ... | ... | ... | ... | ... | ·0051 |
| ,, 7–8 ft. p. min., moderate load, | ... | 0·094 | 0·072 | ... | ... | ·064 |
| ,, ,, ,, heavy ,, ,, | ... | 0·051 | 0·046 | ... | ... | ·047 |
| Railway axles and bearings, (Stahl) | ... | ... | 0·100 | 0·067 | ... | ... |
| Cast-iron steel tyres, . . (Gl) | 0·36 to ·048 | 5 to 30 miles per hour. |  |  |  |  |
| Steel tyres on steel rails, . ,, | 0·11 to 0·04 | 10 to 50 ,, ,, |  |  |  |  |

*Note.*—Rk stands for Rankine; Rn, for Rennie; S, for Summers; W, for Wood; G, for Goodman; RHS, for Prof. Smith; Gl for Gallon and Westinghouse.

## Table CLXXIII.—Conductivity of Sundry Materials.

| | Relative Conductivity at 55° F. | | Conducts Sound Feet per Second. |
|---|---|---|---|
| | Electric. | Thermal. | |
| Silver, annealed | 100·0 | 100·0 | ... |
| Silver, hard drawn | 92·1 | ... | ... |
| Copper, annealed | 94·0 | 91·8 | ... |
| Copper, hard drawn | 91·3 | ... | 11,300 |
| Gold, hard drawn | 71·8 | 53·2 | ... |
| Aluminium, annealed | 51·6 | 31·3 | ... |
| Zinc, pressed | 26·7 | 28·1 | ... |
| Brass tubes (70 cu. 30 zu.) | 22·0 | 26·0 | ... |
| Platinum, annealed | 16·6 | 8·4 | ... |
| Iron and mild steel, annealed | 15·5 | 11·9 | 16,700 |
| Nickel, annealed | 12·05 | 14·0 | ... |
| Tin, pressed | 7·65 | 8·5 | ... |
| German silver | 6·00 | 6·0 | ... |
| Bismuth | 2·00 | 1·8 | ... |
| Mercury, liquid | 1·58 | 1·3 | ... |

## Table CLXXIV.—Various Gases, Properties of.

| Name of Gas. | Density. | Molecular Weight. | No. of Lbs. per Cub. Ft. | No. of Cub. Ft. per Lb. | Air Required per Lb. Gas. | Temperature of Combustion. | Heat Generated by 1 Lb. | Specific Heat Vol. Const. |
|---|---|---|---|---|---|---|---|---|
| | | | | Cub.Ft. | Lbs. | F.° | B.T.U. | |
| Hydrogen | 1·0 | 2 | ·0056 | 178·83 | 34·79 | 4813 | 52,290 | 2·415 |
| Oxygen | 16·0 | 32 | ·0893 | 11·21 | ... | ... | ... | 0·156 |
| Steam | 9·0 | 18 | ·0502 | 19·91 | ... | ... | ... | 0·305 |
| Nitrogen | 14·0 | 28 | ·0785 | 12·76 | ... | ... | ... | 0·174 |
| Air | 14·4 | ... | ·0807 | 12·39 | ... | ... | ... | 0·169 |
| Carbon vapour | 12·0 | ... | ·0670 | 14·93 | 11·59 | 4988 | 20,461 | ... |
| ,, monoxide | 14·0 | 28 | ·0782 | 12·80 | 2·48 | 3494 | 4,383 | 0·173 |
| ,, bisulphide | 38·0 | 76 | ·2124 | 4·71 | 5·48 | 6590 | 9,344 | 0·130 |
| Benzine | 39·0 | 78 | ·2080 | 4·81 | 13·38 | 5022 | 18,000 | 0·350 |
| Acetylene | 13·0 | 26 | ·0727 | 13·46 | 13·38 | 6120 | 21,856 | ... |
| Coal gas | 4·7 | ... | ·0320 | 31·60 | 13·89 | 4500 | 21,400 | 0·468 |
| Blast furnace | 14·0 | ... | ·0790 | 12·65 | 1·00 | 2160 | 1,230 | ... |
| Ammonia | 8·50 | 17 | ·0532 | 18·78 | 6·13 | 3760 | 9,670 | 0·391 |

## Table CLXXV.—Pressure of Water due to various Heads.

| Depth of Water | Pressure in | | Depth of Water. | Pressure in | | Depth of Water. | Pressure in | |
|---|---|---|---|---|---|---|---|---|
| | Pounds per sq. in. | Kilos. per sq. c/m. | | Pounds per sq. in. | Kilos. per sq. c/m. | | Pounds per sq. in. | Kilos. per sq. c/m. |
| 1 in. | ·03608 | ·002537 | 27ft. | 11·691 | ·82196 | 64ft. | 27·712 | 1·94836 |
| 2 | ·07216 | ·005074 | 28 | 12·124 | ·85240 | 65 | 28·145 | 1·97880 |
| 3 | ·10824 | ·007611 | 29 | 12·557 | ·88284 | 66 | 28·578 | 2·00925 |
| 4 | ·14432 | ·010148 | 30 | 12·990 | ·91329 | 67 | 29·011 | 2·03969 |
| 5 | ·18040 | ·012685 | 31 | 13·423 | ·94373 | 68 | 29·444 | 2·07013 |
| 6 | ·21648 | ·015222 | 32 | 13·856 | ·97417 | 69 | 29·877 | 2·10057 |
| 7 | ·25256 | ·017759 | 33 | 14·289 | 1·00462 | 70 | 30·310 | 2·13102 |
| 8 | ·28864 | ·020296 | 34 | 14·722 | 1·03406 | 71 | 30·743 | 2·16146 |
| 9 | ·32472 | ·022833 | 35 | 15·155 | 1·06450 | 72 | 31·176 | 2·19190 |
| 10 | ·36080 | ·025370 | 36 | 15·588 | 1·09495 | 73 | 31·609 | 2·22235 |
| 11 | ·39688 | ·027907 | 37 | 16·021 | 1·12539 | 74 | 32·042 | 2·25279 |
| 1 ft. | ·433 | ·030443 | 38 | 16·454 | 1·15583 | 75 | 32·475 | 2·28323 |
| 2 | ·866 | ·060886 | 39 | 16·887 | 1·18627 | 76 | 32·908 | 2·31368 |
| 3 | 1·299 | ·091329 | 40 | 17·320 | 1·21773 | 77 | 33·341 | 2·34412 |
| 4 | 1·732 | ·121773 | 41 | 17·753 | 1·24817 | 78 | 33·774 | 2·37456 |
| 5 | 2·165 | ·152216 | 42 | 18·186 | 1·27861 | 79 | 34·207 | 2·40500 |
| 6 | 2·598 | ·182659 | 43 | 18·619 | 1·30906 | 80 | 34·640 | 2·43545 |
| 7 | 3·031 | ·213102 | 44 | 19·052 | 1·33950 | 81 | 35·073 | 2·46589 |
| 8 | 3·464 | ·243545 | 45 | 19·485 | 1·36994 | 82 | 35·506 | 2·49633 |
| 9 | 3·897 | ·273989 | 46 | 19·918 | 1·40039 | 83 | 35·939 | 2·52678 |
| 10 | 4·330 | ·30443 | 47 | 20·351 | 1·43083 | 84 | 36·372 | 2·55722 |
| 11 | 4·763 | ·33487 | 48 | 20·784 | 1·46127 | 85 | 36·805 | 2·58766 |
| 12 | 5·196 | ·36531 | 49 | 21·217 | 1·49171 | 86 | 37·238 | 2·61811 |
| 13 | 5·629 | ·39576 | 50 | 21·650 | 1·52216 | 87 | 37·671 | 2·64855 |
| 14 | 6·062 | ·42620 | 51 | 22·083 | 1·55260 | 88 | 38·104 | 2·67899 |
| 15 | 6·495 | ·45664 | 52 | 22·516 | 1·58304 | 89 | 38·537 | 2·70943 |
| 16 | 6·928 | ·48709 | 53 | 22·949 | 1·61349 | 90 | 38·970 | 2·73989 |
| 17 | 7·361 | ·51753 | 54 | 23·382 | 1·64393 | 91 | 39·403 | 2·77033 |
| 18 | 7·794 | ·54797 | 55 | 23·815 | 1·67437 | 92 | 39·836 | 2·80077 |
| 19 | 8·227 | ·57841 | 56 | 24·248 | 1·70482 | 93 | 40·269 | 2·83122 |
| 20 | 8·660 | ·60886 | 57 | 24·681 | 1·73526 | 94 | 40·702 | 2·86166 |
| 21 | 9·093 | ·63930 | 58 | 25·114 | 1·76570 | 95 | 41·135 | 2·89210 |
| 22 | 9·526 | ·66974 | 59 | 25·557 | 1·79614 | 96 | 41·568 | 2·92255 |
| 23 | 9·959 | ·70019 | 60 | 25·980 | 1·82659 | 97 | 42·001 | 2·95299 |
| 24 | 10·392 | ·73063 | 61 | 26·413 | 1·85703 | 98 | 42·434 | 2·98343 |
| 25 | 10·825 | ·76107 | 62 | 26·846 | 1·88747 | 99 | 42·867 | 3·01387 |
| 26 | 11·258 | ·79152 | 63 | 27·279 | 1·91792 | 100 | 43·300 | 3·04432 |

**The above Table is calculated for fresh water at a temperature of 62° F**

## Table CLXXVI.—Coal Consumption in Tons per Day.

| Indicated Horse-Power. | Pounds per Indicated Horse-Power per Hour. | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 1·0 | 1·1 | 1·2 | 1·3 | 1·4 | 1·5 | 1·6 | 1·7 | 1·8 | 1·9 | 2·0 | 2·1 | 2·2 | 2·3 | 2·4 | 2·5 |
| 10,000 | 107·1 | 117·8 | 128·5 | 139·3 | 150·0 | 160·7 | 171·4 | 182·1 | 192·8 | 203·5 | 214·2 | 225·0 | 235·7 | 246·4 | 257·1 | 267·8 |
| 9,000 | 96·4 | 106·0 | 115·7 | 125·3 | 135·0 | 144·6 | 154·2 | 163·9 | 173·5 | 183·2 | 192·7 | 202·5 | 212·1 | 221·7 | 231·4 | 241·0 |
| 8,000 | 85·7 | 94·2 | 102·8 | 111·4 | 120·0 | 128·5 | 137·1 | 145·7 | 154·2 | 162·4 | 171·4 | 180·0 | 188·5 | 197·1 | 205·7 | 214·2 |
| 7,000 | 75·0 | 82·5 | 90·0 | 97·5 | 105·0 | 112·5 | 120·0 | 127·5 | 135·0 | 142·5 | 150·0 | 157·5 | 165·0 | 172·5 | 180·0 | 187·5 |
| 6,000 | 64·3 | 70·7 | 77·1 | 83·5 | 90·0 | 96·4 | 102·8 | 109·2 | 115·7 | 122·1 | 128·5 | 135·0 | 141·4 | 147·8 | 154·2 | 160·7 |
| 5,000 | 53·5 | 58·9 | 64·2 | 69·6 | 75·0 | 80·3 | 85·7 | 91·0 | 96·4 | 101·8 | 107·1 | 112·5 | 117·8 | 123·2 | 128·5 | 134·0 |
| 4,000 | 42·8 | 47·1 | 51·4 | 55·7 | 60·0 | 64·2 | 68·5 | 72·8 | 77·1 | 81·4 | 85·7 | 90·0 | 94·2 | 98·5 | 102·8 | 107·1 |
| 3,000 | 32·1 | 35·3 | 38·6 | 41·7 | 45·0 | 48·2 | 51·4 | 54·6 | 57·9 | 61·0 | 64·2 | 67·5 | 70·7 | 73·9 | 77·1 | 80·3 |
| 2,000 | 21·4 | 23·5 | 25·7 | 27·8 | 30·0 | 32·1 | 34·2 | 36·4 | 38·5 | 41·7 | 42·8 | 45·0 | 47·1 | 49·2 | 51·4 | 53·5 |
| 1,000 | 10·7 | 11·8 | 12·8 | 13·9 | 15·0 | 16·1 | 17·1 | 18·2 | 19·3 | 20·3 | 21·4 | 22·5 | 23·6 | 24·6 | 25·7 | 26·8 |
| 900 | 9·64 | 10·62 | 11·57 | 12·53 | 13·50 | 14·46 | 15·42 | 16·39 | 17·35 | 18·32 | 19·27 | 20·25 | 21·21 | 22·17 | 23·14 | 24·10 |
| 800 | 8·57 | 9·42 | 10·28 | 11·14 | 12·00 | 12·85 | 13·71 | 14·57 | 15·42 | 16·24 | 17·14 | 18·00 | 18·85 | 19·71 | 20·57 | 21·42 |
| 700 | 7·50 | 8·25 | 9·00 | 9·75 | 10·50 | 11·25 | 12·00 | 12·75 | 13·50 | 14·25 | 15·00 | 15·75 | 16·50 | 17·25 | 18·00 | 18·75 |
| 600 | 6·43 | 7·07 | 7·71 | 8·35 | 9·00 | 9·64 | 10·28 | 10·92 | 11·57 | 12·21 | 12·85 | 13·50 | 14·14 | 14·78 | 15·42 | 16·07 |
| 500 | 5·35 | 5·89 | 6·42 | 6·96 | 7·50 | 8·03 | 8·57 | 9·10 | 9·64 | 10·18 | 10·71 | 11·25 | 11·78 | 12·32 | 12·85 | 13·40 |
| 400 | 4·28 | 4·71 | 5·14 | 5·57 | 6·00 | 6·42 | 6·85 | 7·28 | 7·71 | 8·14 | 8·57 | 9·00 | 9·42 | 9·85 | 10·28 | 10·71 |
| 300 | 3·21 | 3·53 | 3·86 | 4·17 | 4·50 | 4·82 | 5·14 | 5·46 | 5·78 | 6·10 | 6·42 | 6·75 | 7·07 | 7·39 | 7·71 | 8·03 |
| 200 | 2·14 | 2·35 | 2·57 | 2·78 | 3·00 | 3·21 | 3·42 | 3·64 | 3·85 | 4·07 | 4·28 | 4·50 | 4·71 | 4·92 | 5·14 | 5·35 |
| 100 | 1·07 | 1·18 | 1·28 | 1·89 | 1·50 | 1·61 | 1·71 | 1·82 | 1·92 | 2·03 | 2·14 | 2·25 | 2·35 | 2·46 | 2·57 | 2·68 |
| 90 | ·964 | 1·06 | 1·16 | 1·25 | 1·35 | 1·44 | 1·54 | 1·64 | 1·73 | 1·83 | 1·93 | 2·02 | 2·12 | 2·22 | 2·31 | 2·41 |
| 80 | ·857 | ·942 | 1·03 | 1·11 | 1·20 | 1·28 | 1·37 | 1·46 | 1·54 | 1·62 | 1·71 | 1·80 | 1·88 | 1·97 | 2·06 | 2·14 |
| 70 | ·750 | ·825 | ·900 | ·975 | 1·05 | 1·12 | 1·20 | 1·27 | 1·35 | 1·42 | 1·50 | 1·57 | 1·65 | 1·72 | 1·80 | 1·88 |
| 60 | ·643 | ·707 | ·771 | ·835 | ·900 | ·964 | 1·03 | 1·09 | 1·15 | 1·22 | 1·28 | 1·35 | 1·41 | 1·48 | 1·54 | 1·61 |
| 50 | ·535 | ·589 | ·642 | ·696 | ·750 | ·803 | ·857 | ·910 | ·964 | 1·01 | 1·07 | 1·12 | 1·17 | 1·23 | 1·29 | 1·34 |
| 40 | ·428 | ·471 | ·514 | ·557 | ·600 | ·642 | ·685 | ·728 | ·771 | ·814 | ·857 | ·900 | ·942 | ·985 | 1·03 | 1·07 |
| 30 | ·321 | ·353 | ·386 | ·417 | ·450 | ·482 | ·514 | ·546 | ·578 | ·610 | ·642 | ·675 | ·707 | ·739 | ·771 | ·803 |
| 20 | ·214 | ·235 | ·257 | ·278 | ·300 | ·321 | ·342 | ·364 | ·385 | ·407 | ·428 | ·450 | ·471 | ·492 | ·514 | ·536 |
| 10 | ·107 | ·118 | ·128 | ·139 | ·150 | ·161 | ·171 | ·182 | ·192 | ·203 | ·214 | ·225 | ·235 | ·246 | ·257 | ·268 |

## Table CLXXVII.—Consumption of Oil and other Fuels in tons per 24 hours.

| Horse Power. | Consumption per Horse-power hour in fractions of a pound. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0·25 | 0·30 | 0·35 | 0·40 | 0·45 | 0·50 | 0·55 | 0·60 | 0·65 | 0·70 | 0·75 | 0·80 | 0·85 | 0·90 |
| 100 | 0·269 | 0·323 | 0·376 | 0·430 | 0·484 | 0·538 | 0·591 | 0·645 | 0·699 | 0·753 | 0·806 | 0·860 | 0·914 | 0·968 |
| 150 | 0·403 | 0·484 | 0·564 | 0·645 | 0·726 | 0·806 | 0·887 | 0·968 | 1·048 | 1·129 | 1·209 | 1·290 | 1·371 | 1·451 |
| 200 | 0·538 | 0·645 | 0·753 | 0·860 | 0·967 | 1·075 | 1·183 | 1·290 | 1·398 | 1·505 | 1·613 | 1·720 | 1·828 | 1·935 |
| 250 | 0·672 | 0·806 | 0·941 | 1·075 | 1·209 | 1·344 | 1·478 | 1·613 | 1·747 | 1·881 | 2·016 | 2·150 | 2·284 | 2·419 |
| 300 | 0·806 | 0·968 | 1·129 | 1·290 | 1·451 | 1·613 | 1·774 | 1·935 | 2·096 | 2·258 | 2·418 | 2·580 | 2·741 | 2·903 |
| 350 | 0·941 | 1·129 | 1·317 | 1·505 | 1·693 | 1·881 | 2·069 | 2·258 | 2·446 | 2·634 | 2·821 | 3·070 | 3·198 | 3·386 |
| 400 | 1·075 | 1·290 | 1·505 | 1·720 | 1·935 | 2·150 | 2·365 | 2·580 | 2·795 | 3·010 | 3·224 | 3·440 | 3·655 | 3·870 |
| 450 | 1·210 | 1·451 | 1·693 | 1·935 | 2·177 | 2·419 | 2·661 | 2·903 | 3·144 | 3·386 | 3·628 | 3·870 | 4·112 | 4·354 |
| 500 | 1·344 | 1·613 | 1·881 | 2·150 | 2·419 | 2·688 | 2·956 | 3·225 | 3·494 | 3·763 | 4·031 | 4·300 | 4·569 | 4·838 |
| 600 | 1·613 | 1·935 | 2·258 | 2·580 | 2·902 | 3·225 | 3·547 | 3·870 | 4·193 | 4·515 | 4·837 | 5·160 | 5·483 | 5·805 |
| 700 | 1·882 | 2·258 | 2·634 | 3·010 | 3·386 | 3·763 | 4·139 | 4·515 | 4·891 | 5·268 | 5·643 | 6·020 | 6·396 | 6·773 |
| 800 | 2·150 | 2·580 | 3·010 | 3·440 | 3·870 | 4·300 | 4·730 | 5·160 | 5·590 | 6·020 | 6·449 | 6·880 | 7·310 | 7·740 |
| 900 | 2·419 | 2·901 | 3·386 | 3·870 | 4·354 | 4·838 | 5·321 | 5·805 | 6·289 | 6·773 | 7·255 | 7·740 | 8·224 | 8·708 |
| 1000 | 2·688 | 3·225 | 3·763 | 4·300 | 4·837 | 5·375 | 5·812 | 6·450 | 6·988 | 7·525 | 8·061 | 8·600 | 9·138 | 9·675 |
| 1250 | 3·360 | 4·031 | 4·703 | 5·375 | 6·047 | 6·719 | 7·266 | 8·063 | 8·734 | 9·406 | 10·08 | 10·75 | 11·42 | 12·09 |
| 1500 | 4·032 | 4·838 | 5·644 | 6·450 | 7·256 | 8·063 | 8·719 | 9·675 | 10·81 | 11·29 | 12·09 | 12·90 | 13·71 | 14·51 |
| 1750 | 4·704 | 5·644 | 6·584 | 7·525 | 8·465 | 9·406 | 10·17 | 11·29 | 12·23 | 13·17 | 14·11 | 15·05 | 15·99 | 16·93 |
| 2000 | 5·376 | 6·450 | 7·525 | 8·600 | 9·675 | 10·75 | 11·62 | 12·90 | 13·98 | 15·05 | 16·12 | 17·20 | 18·28 | 19·35 |
| 2500 | 6·720 | 8·063 | 9·406 | 10·75 | 12·09 | 13·44 | 14·53 | 16·13 | 17·47 | 18·81 | 20·15 | 21·50 | 22·84 | 24·19 |
| 3000 | 8·064 | 9·675 | 11·29 | 12·90 | 14·51 | 16·12 | 17·44 | 19·45 | 20·96 | 22·57 | 24·19 | 25·80 | 27·41 | 29·03 |
| 3500 | 9·408 | 11·29 | 13·17 | 15·05 | 16·93 | 18·81 | 20·34 | 22·68 | 24·46 | 26·33 | 28·22 | 30·10 | 31·98 | 33·87 |
| 4000 | 10·75 | 12·90 | 15·05 | 17·20 | 19·35 | 21·50 | 23·25 | 25·90 | 27·95 | 30·10 | 32·25 | 34·40 | 36·55 | 38·71 |
| 5000 | 13·44 | 16·13 | 18·81 | 21·50 | 24·19 | 26·87 | 29·16 | 32·22 | 34·94 | 37·61 | 40·30 | 43·00 | 45·69 | 48·39 |

## Table CLXXVIII.—Comparison of Thermometers.

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. |
|---|---|---|---|---|---|---|---|---|
| −461·20 | −274·00 | −219·22 | + | − | − | + | + | + |
| + | − | − | 21·20 | 6· | 4·80 | 44·60 | 7· | 5·60 |
| 0· | 17·78 | 14·22 | 22· | 5·56 | 4·44 | 45· | 7·23 | 5·77 |
| 0·50 | 17·50 | 14· | 23· | 5· | 4· | 45·50 | 7·50 | 6· |
| 1· | 17·23 | 13·77 | 24· | 4·45 | 3·55 | 46· | 7·78 | 6·22 |
| 1·40 | 17· | 13·60 | 24·80 | 4· | 3·20 | 46·40 | 8· | 6·40 |
| 2· | 16·67 | 13·33 | 25· | 3·89 | 3·11 | 47· | 8·34 | 6·66 |
| 2·75 | 16·25 | 13· | 25·25 | 3·75 | 3· | 47·75 | 8·75 | 7· |
| 3· | 16·11 | 12·88 | 26· | 3·34 | 2·66 | 48· | 8·89 | 7·11 |
| 3·20 | 16· | 12·80 | 26·60 | 3· | 2·40 | 48·20 | 9· | 7·20 |
| 4· | 15·56 | 12·44 | 27· | 2·78 | 2·22 | 49· | 9·45 | 7·55 |
| 5· | 15· | 12· | 27·50 | 2·50 | 2· | 50· | 10· | 8· |
| 6· | 14·45 | 11·55 | 28· | 2·23 | 1·77 | 51· | 10·56 | 8·44 |
| 6·80 | 14· | 11·20 | 28·40 | 2· | 1·60 | 51·80 | 11· | 8·80 |
| 7· | 13·89 | 11·11 | 29· | 1·67 | 1·33 | 52· | 11·11 | 8·88 |
| 7·25 | 13·75 | 11· | 29·75 | 1·25 | 1· | 52·25 | 11·25 | 9· |
| 8· | 13·34 | 10·66 | 30· | 1·11 | 0·88 | 53· | 11·67 | 9·33 |
| 8·60 | 13· | 10·40 | 30·20 | 1· | 0·80 | 53·60 | 12· | 9·60 |
| 9· | 12·78 | 10·22 | 31· | 0·56 | 0·44 | 54· | 12·23 | 9·77 |
| 9·50 | 12·50 | 10· | 32· | 0· | 0· | 54·50 | 12·50 | 10· |
| 10· | 12·23 | 9·77 | +33· | +0·56 | +0·44 | 55· | 12·78 | 10·22 |
| 10·40 | 12· | 9·60 | 33·80 | 1· | 0·80 | 55·40 | 13· | 10·40 |
| 11· | 11·67 | 9·33 | 34· | 1·11 | 0·88 | 56· | 13·34 | 10·66 |
| 11·75 | 11·25 | 9· | 34·25 | 1·25 | 1· | 56·75 | 13·75 | 11· |
| 12· | 11·11 | 8·88 | 35· | 1·67 | 1·33 | 57· | 13·89 | 11·11 |
| 12·20 | 11· | 8·80 | 35·60 | 2· | 1·60 | 57·20 | 14· | 11·20 |
| 13· | 10·56 | 8·44 | 36· | 2·23 | 1·77 | 58· | 14·45 | 11·55 |
| 14· | 10· | 8· | 36·50 | 2·50 | 2· | 59· | 15· | 12· |
| 15· | 9·45 | 7·55 | 37· | 2·78 | 2·22 | 60· | 15·56 | 12·44 |
| 15·80 | 9· | 7·20 | 37·40 | 3· | 2·40 | 60·80 | 16· | 12·80 |
| 16· | 8·89 | 7·11 | 38· | 3·34 | 2·66 | 61· | 16·11 | 12·88 |
| 16·25 | 8·75 | 7· | 38·75 | 3·75 | 3· | 61·25 | 16·25 | 13· |
| 17· | 8·34 | 6·66 | 39· | 3·89 | 3·11 | 62· | 16·67 | 13·33 |
| 17·60 | 8· | 6·40 | 39·20 | 4· | 3·20 | 62·60 | 17· | 13·60 |
| 18· | 7·78 | 6·22 | 40· | 4·45 | 3·55 | 63· | 17·23 | 13·77 |
| 18·50 | 7·50 | 6· | 41· | 5· | 4· | 63·50 | 17·50 | 14· |
| 19· | 7·23 | 5·77 | 42· | 5·56 | 4·44 | 64· | 17·78 | 14·22 |
| 19·40 | 7· | 5·60 | 42·80 | 6· | 4·80 | 64·40 | 18· | 14·40 |
| 20· | 6·67 | 5·33 | 43· | 6·11 | 4·88 | 65· | 18·34 | 14·66 |
| 20·75 | 6·25 | 5· | 43·25 | 6·25 | 5· | 65·75 | 18·75 | 15· |
| 21· | 6·11 | 4·88 | 44· | 6·67 | 5·33 | 66· | 18·89 | 15·11 |

***Note.*—Temp. Fahr. = $\frac{9}{5}$ temp. Cent. + 32°.**
**Temp. Cent. = $\frac{5}{9}$ (temp Fahr. − 32°).**

Table CLXXVIII.—Comparison of Thermometers—*continued.*

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. |
|---|---|---|---|---|---|---|---|---|
| + | + | + | + | + | + | + | + | + |
| 66·20 | 19· | 15·20 | 89·60 | 32· | 25·60 | 111·20 | 44· | 35·20 |
| 67· | 19·45 | 15·55 | 90· | 32·23 | 25·77 | 112· | 44·45 | 35·55 |
| 68· | 20· | 16· | 90·50 | 32·50 | 26· | 113· | 45· | 36· |
| 69· | 20·56 | 16·44 | 91· | 32·78 | 26·22 | 114· | 45·56 | 36·44 |
| 69·80 | 21· | 16·80 | 91·40 | 33· | 26·40 | 114·80 | 46· | 36·80 |
| 70· | 21·11 | 16·88 | 92· | 33·34 | 26·66 | 115· | 46·11 | 36·88 |
| 70·25 | 21·25 | 17· | 92·75 | 33·75 | 27· | 115·25 | 46·25 | 37· |
| 71· | 21·67 | 17·33 | 93· | 33·89 | 27·11 | 116· | 46·67 | 37·33 |
| 71·60 | 22· | 17·60 | 93·20 | 34· | 27·20 | 116·60 | 47· | 37·60 |
| 72· | 22·23 | 17·77 | 94· | 34·45 | 27·55 | 117· | 47·23 | 37·77 |
| 72·50 | 22·50 | 18· | 95· | 35· | 28· | 117·50 | 47·50 | 38· |
| 73· | 22·78 | 18·22 | 96· | 35·56 | 28·44 | 118· | 47·78 | 38·22 |
| 73·40 | 23· | 18·40 | 96·80 | 36· | 28·80 | 118·40 | 48· | 38·40 |
| 74· | 23·34 | 18·66 | 97· | 36·11 | 28·88 | 119· | 48·34 | 38·66 |
| 74·75 | 23·75 | 19· | 97·25 | 36·25 | 29· | 119·75 | 48·75 | 39· |
| 75· | 23·89 | 19·11 | 98· | 36·67 | 29·33 | 120· | 48·89 | 39·11 |
| 75·20 | 24· | 19·20 | 98·60 | 37· | 29·60 | 120·20 | 49· | 39·20 |
| 76· | 24·45 | 19·55 | 99· | 37·23 | 29·77 | 121· | 49·45 | 39·55 |
| 77· | 25· | 20· | 99·50 | 37·50 | 30· | 122· | 50· | 40· |
| 78· | 25·56 | 20·44 | 100· | 37·78 | 30·22 | 123· | 50·56 | 40·44 |
| 78·80 | 26· | 20·80 | 100·40 | 38· | 30·40 | 123·80 | 51· | 40·80 |
| 79· | 26·11 | 20·88 | 101· | 38·34 | 30·66 | 124· | 51·11 | 40·88 |
| 79·25 | 26·25 | 21· | 101·75 | 38·75 | 31· | 124·25 | 51·25 | 41· |
| 80· | 26·67 | 21·33 | 102· | 38·89 | 31·11 | 125· | 51·67 | 41·33 |
| 80·60 | 27· | 21·60 | 102·20 | 39· | 31·20 | 125·60 | 52· | 41·60 |
| 81· | 27·23 | 21·77 | 103· | 39·45 | 31·55 | 126· | 52·23 | 41·77 |
| 81·50 | 27·50 | 22· | 104· | 40· | 32· | 126·50 | 52·50 | 42· |
| 82· | 27·78 | 22·22 | 105· | 40·56 | 32·44 | 127· | 52·78 | 42·22 |
| 82·40 | 28· | 22·40 | 105·80 | 41· | 32·80 | 127·40 | 53· | 42·40 |
| 83· | 28·34 | 22·66 | 106· | 41·11 | 32·88 | 128· | 53·34 | 42·66 |
| 83·75 | 28·75 | 23· | 106·25 | 41·25 | 33· | 128·75 | 53·75 | 43· |
| 84· | 28·89 | 23·11 | 107· | 41·67 | 33·33 | 129· | 53·89 | 43·11 |
| 84·20 | 29· | 23·20 | 107·60 | 42· | 33·60 | 129·20 | 54· | 43·20 |
| 85· | 29·45 | 23·55 | 108· | 42·23 | 33·77 | 130· | 54·45 | 43·55 |
| 86· | 30· | 24· | 108·50 | 42·50 | 34· | 131· | 55· | 44· |
| 87· | 30·56 | 24·44 | 109· | 42·78 | 34·22 | 132· | 55·56 | 44·44 |
| 87·80 | 31· | 24·80 | 109·40 | 43· | 34·40 | 132·80 | 56· | 44·80 |
| 88· | 31·11 | 24·88 | 110· | 43·34 | 34·66 | 133· | 56·11 | 44·88 |
| 88·25 | 31·25 | 25· | 110·75 | 43·75 | 35· | 133·25 | 56·25 | 45· |
| 89· | 31·67 | 25·33 | 111· | 43·89 | 35·11 | 134· | 56·67 | 45·33 |

## Table CLXXVIII.—Comparison of Thermometers—*continued.*

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. |
|---|---|---|---|---|---|---|---|---|
| + | + | + | + | + | + | + | + | + |
| 134·60 | 57· | 45·60 | 156·20 | 69· | 55·20 | 179·60 | 82· | 65·60 |
| 135· | 57·23 | 45·77 | 157· | 69·45 | 55·55 | 180· | 82·23 | 65·77 |
| 135·50 | 57·50 | 46· | 158· | 70· | 56· | 180·50 | 82·50 | 66· |
| 136· | 57·78 | 46·22 | 159· | 70·56 | 56·44 | 181· | 82·78 | 66·22 |
| 136·40 | 58· | 46·40 | 159·80 | 71· | 56·80 | 181·40 | 83· | 66·40 |
| 137· | 58·34 | 46·66 | 160· | 71·11 | 56·88 | 182· | 83·34 | 66·66 |
| 137·75 | 58·75 | 47· | 160·25 | 71·25 | 57· | 182·75 | 83·75 | 67· |
| 138· | 58·89 | 47 11 | 161· | 71·67 | 57·33 | 183· | 83·89 | 67·11 |
| 138·20 | 59· | 47·20 | 161·60 | 72· | 57·60 | 183·20 | 84· | 67·20 |
| 139· | 59·45 | 47·55 | 162· | 72·23 | 57·77 | 184· | 84·45 | 67·55 |
| 140· | 60· | 48· | 162·50 | 72·50 | 58· | 185· | 85· | 68· |
| 141· | 60·56 | 48·44 | 163· | 72·78 | 58·22 | 186· | 85·56 | 68·44 |
| 141·80 | 61· | 48·80 | 163·40 | 73· | 58·40 | 186·80 | 86· | 68·80 |
| 142· | 61·11 | 48·88 | 164· | 73·34 | 58·66 | 187· | 86·11 | 68·88 |
| 142·25 | 61·25 | 49· | 164·75 | 73·75 | 59· | 187·25 | 86·25 | 69· |
| 143· | 61·67 | 49·33 | 165· | 73·89 | 59·11 | 188· | 86·67 | 69·33 |
| 143·60 | 62· | 49·60 | 165·20 | 74· | 59·20 | 188·60 | 87· | 69·60 |
| 144· | 62·23 | 49·77 | 166· | 74·45 | 59·55 | 189· | 87·23 | 69·77 |
| 144·50 | 62·50 | 50· | 167· | 75· | 60· | 189·50 | 87·50 | 70· |
| 145· | 62·78 | 50·22 | 168· | 75·56 | 60·44 | 190· | 87·78 | 70·22 |
| 145·40 | 63· | 50·40 | 168·80 | 76· | 60·80 | 190·40 | 88· | 70·40 |
| 146· | 63·34 | 50·66 | 169· | 76·11 | 60·88 | 191· | 88·34 | 70·66 |
| 146·75 | 63·75 | 51· | 169·25 | 76·25 | 61· | 191·75 | 88·75 | 71· |
| 147· | 63·89 | 51·11 | 170· | 76·67 | 61·33 | 192· | 88·89 | 71·11 |
| 147·20 | 54· | 51·20 | 170·60 | 77· | 61·60 | 192·20 | 89· | 71·20 |
| 148· | 64·45 | 51·55 | 171· | 77·23 | 61·77 | 193· | 89·45 | 71·55 |
| 149· | 65· | 52· | 171·50 | 77·50 | 62· | 194· | 90· | 72· |
| 150· | 65·56 | 52·44 | 172· | 77·78 | 62·22 | 195· | 90·56 | 72·44 |
| 150·80 | 66· | 52·80 | 172·40 | 78· | 62·40 | 195·80 | 91· | 72·80 |
| 151· | 66·11 | 52·88 | 173· | 78·34 | 62·66 | 196· | 91·11 | 72·88 |
| 151·25 | 66·25 | 53· | 173·75 | 78·75 | 63· | 196·25 | 91·25 | 73· |
| 152· | 66·67 | 53·33 | 174· | 78·89 | 63·11 | 197· | 91·67 | 73·33 |
| 152·60 | 67· | 53·60 | 174·20 | 79· | 63·20 | 197·60 | 92· | 73·60 |
| 153· | 67·23 | 53·77 | 175· | 79·45 | 63·55 | 198· | 92·23 | 73·77 |
| 153·50 | 67·50 | 54· | 176· | 80· | 64· | 198·50 | 92·50 | 74· |
| 154· | 67·78 | 54·22 | 177· | 80·56 | 64·44 | 199· | 92·78 | 74·22 |
| 154·40 | 68· | 54·40 | 177·80 | 81· | 64·80 | 199·40 | 93· | 74·40 |
| 155· | 68·34 | 54·66 | 178· | 81·11 | 64·88 | 200· | 93·34 | 74·66 |
| 155·75 | 68·75 | 55· | 178·25 | 81·25 | 65· | 200·75 | 93·75 | 75· |
| 156· | 68·89 | 55·11 | 179· | 81·67 | 65·33 | 201· | 93·89 | 75·11 |

Table CLXXVIII.—Comparison of Thermometers—*continued.*

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. |
|---|---|---|---|---|---|---|---|---|
| + | + | + | + | + | + | + | + | + |
| 201·20 | 94· | 75·20 | 224·60 | 107· | 85·60 | 246·20 | 119· | 95·20 |
| 202· | 94·45 | 75·55 | 225· | 107·23 | 85·77 | 247· | 119·45 | 95·55 |
| 203· | 95· | 76· | 225·50 | 107·50 | 86· | 248· | 120· | 96· |
| 204· | 95·56 | 76·44 | 226· | 107·78 | 86·22 | 249· | 120·56 | 96·44 |
| 204·80 | 96· | 76·80 | 226·40 | 108· | 86·40 | 249·80 | 121· | 96·80 |
| 205· | 96·11 | 76·88 | 227· | 108·34 | 86·66 | 250· | 121·11 | 96·88 |
| 205·25 | 96·25 | 77· | 227·75 | 108·75 | 87· | 250·25 | 121·25 | 97· |
| 206· | 96·67 | 77·33 | 228· | 108·89 | 87·11 | 251· | 121·67 | 97·33 |
| 206·60 | 97· | 77·60 | 228·20 | 109· | 87·20 | 251·60 | 122· | 97·60 |
| 207· | 97·23 | 77·77 | 229· | 109·45 | 87·55 | 252· | 122·23 | 97·77 |
| 207·50 | 97·50 | 78· | 230· | 110· | 88· | 252·50 | 122·50 | 98· |
| 208· | 97·78 | 78·22 | 231· | 110·56 | 88·44 | 253· | 122·78 | 98·22 |
| 208·40 | 98· | 78·40 | 231·80 | 111· | 88·80 | 253·40 | 123· | 98·40 |
| 209· | 98·34 | 78·66 | 232· | 111·11 | 88·88 | 254· | 123·34 | 98·66 |
| 209·75 | 98·75 | 79· | 232·25 | 111·25 | 89· | 254·75 | 123·75 | 99· |
| 210· | 98·89 | 79·11 | 233· | 111·67 | 89·33 | 255· | 123·89 | 99·11 |
| 210·20 | 99· | 79·20 | 233·60 | 112· | 89·60 | 255·20 | 124· | 99·20 |
| 211· | 99·45 | 79·55 | 234· | 112·23 | 89·77 | 256· | 124·45 | 99·55 |
| 212· | 100· | 80· | 234·50 | 112·50 | 90· | 257· | 125· | 100· |
| 213· | 100·56 | 80·44 | 235· | 112·78 | 90·22 | 258· | 125·56 | 100·44 |
| 213·80 | 101· | 80·80 | 235·40 | 113· | 90·40 | 258·80 | 126· | 100·80 |
| 214· | 101·11 | 80·88 | 236· | 113·34 | 90·66 | 259· | 126·11 | 100·88 |
| 214·25 | 101·25 | 81· | 236·75 | 113·75 | 91· | 259·25 | 126·25 | 101· |
| 215· | 101·67 | 81·33 | 237· | 113·89 | 91·11 | 260· | 126·67 | 101·33 |
| 215·60 | 102· | 81·60 | 237·20 | 114· | 91·20 | 260·60 | 127· | 101·60 |
| 216· | 102·23 | 81·77 | 238· | 114·45 | 91·55 | 261· | 127·23 | 101·77 |
| 216·50 | 102·50 | 82· | 239· | 115· | 92· | 261·50 | 127·50 | 102· |
| 217· | 102·78 | 82·22 | 240· | 115·56 | 92·44 | 262· | 127·78 | 102·22 |
| 217·40 | 103· | 82·40 | 240·80 | 116· | 92·80 | 262·40 | 128· | 102·40 |
| 218· | 103·34 | 82·66 | 241· | 116·11 | 92·88 | 263· | 128·34 | 102·66 |
| 218·75 | 103·75 | 83· | 241·25 | 116·25 | 93· | 263·75 | 128·75 | 103· |
| 219· | 103·89 | 83·11 | 242· | 116·67 | 93·33 | 264· | 128·89 | 103·11 |
| 219·20 | 104· | 83·20 | 242·60 | 117· | 93·60 | 264·20 | 129· | 103·20 |
| 220· | 104·45 | 83·55 | 243· | 117·23 | 93·77 | 265· | 129·45 | 103·55 |
| 221· | 105· | 84· | 243·50 | 117·50 | 94· | 266· | 130· | 104· |
| 222· | 105·56 | 84·44 | 244· | 117·78 | 94·22 | 267· | 130·26 | 104·44 |
| 222·80 | 106· | 84·80 | 244·40 | 118· | 94·40 | 267·80 | 131· | 104·80 |
| 223· | 106·11 | 84·88 | 245· | 118·34 | 94·66 | 268· | 131·11 | 104·88 |
| 223·25 | 106·25 | 85· | 245·75 | 118·75 | 95· | 268·25 | 131·25 | 105· |
| 224· | 106·67 | 85·33 | 246· | 118·89 | 95·11 | 269· | 131·67 | 105·33 |

## Table CLXXVIII.—Comparison of Thermometers—*continued.*

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reau mur degs. |
|---|---|---|---|---|---|---|---|---|
| + | + | + | + | + | + | + | + | + |
| 269·60 | 132· | 105·60 | 291·20 | 144· | 115·20 | 314·60 | 157· | 125·60 |
| 270· | 132·23 | 105·77 | 292· | 144·45 | 115·55 | 315· | 157·23 | 125·77 |
| 270·50 | 132·50 | 106· | 293· | 145· | 116· | 315·50 | 157·50 | 126· |
| 271· | 132·78 | 106·22 | 294· | 145·56 | 116·44 | 316· | 157·78 | 126·22 |
| 271·40 | 133· | 106·40 | 294·80 | 146· | 116·80 | 316·40 | 158· | 126·40 |
| 272· | 133·34 | 106·66 | 295· | 146·11 | 116·88 | 317· | 158·34 | 126·66 |
| 272·75 | 133·75 | 107· | 295·25 | 146·25 | 117· | 317·75 | 158·75 | 127· |
| 273· | 133·89 | 107·11 | 296· | 146·67 | 117·33 | 318· | 158·89 | 127·11 |
| 273·20 | 134· | 107·20 | 296·60 | 147· | 117·60 | 318·20 | 159· | 127·20 |
| 274· | 134·45 | 107·55 | 297· | 147·23 | 117·77 | 319· | 159·45 | 127·55 |
| 275· | 135· | 108· | 297·50 | 147·50 | 118· | 320· | 160· | 128· |
| 276· | 135·56 | 108·44 | 298· | 147·78 | 118·22 | 821· | 160·56 | 128·44 |
| 276·80 | 136· | 108·80 | 298·40 | 148· | 118·40 | 321·80 | 161· | 128·80 |
| 277· | 136·11 | 108·88 | 299· | 148·34 | 118·66 | 322· | 161·11 | 128·88 |
| 277·25 | 136·25 | 109· | 299·75 | 148·75 | 119· | 322·25 | 161·25 | 129· |
| 278· | 136·67 | 109·33 | 300· | 148·89 | 119·11 | 323· | 161·67 | 129·33 |
| 278·60 | 137· | 109·60 | 300·20 | 149· | 119·20 | 323·60 | 162· | 129·60 |
| 279· | 137·23 | 109·77 | 301· | 149·45 | 119·55 | 324· | 162·23 | 129·77 |
| 279·50 | 137·50 | 110· | 302· | 150· | 120· | 324·50 | 162·50 | 130· |
| 280· | 137·78 | 110·22 | 303· | 150·56 | 120·44 | 325· | 162·78 | 130·22 |
| 280·40 | 138· | 110·40 | 303·80 | 151· | 120·80 | 325·40 | 163· | 130·40 |
| 281· | 138·34 | 110·66 | 304· | 151·11 | 120·88 | 326· | 163·34 | 130·66 |
| 281·75 | 138·75 | 111· | 304·25 | 151·25 | 121· | 326·75 | 163·75 | 131· |
| 282· | 138·89 | 111·11 | 305· | 151·67 | 121·33 | 327· | 163·89 | 131·11 |
| 282·20 | 139· | 111·20 | 305·60 | 152· | 121·60 | 327·20 | 164· | 131·20 |
| 283· | 139·45 | 111·55 | 306· | 152·23 | 121·77 | 328· | 164·45 | 131·55 |
| 284· | 140· | 112· | 306·50 | 152·50 | 122· | 329· | 165· | 132· |
| 285· | 140·56 | 112·44 | 307· | 152·78 | 122·22 | 330· | 165·56 | 132·44 |
| 285·80 | 141· | 112·80 | 307·40 | 153· | 122·40 | 330·80 | 166· | 132·80 |
| 286· | 141·11 | 112·88 | 308· | 153·34 | 122·66 | 331· | 166·11 | 132·88 |
| 286·25 | 141·25 | 113· | 308·75 | 153·75 | 123· | 331·25 | 166·25 | 133· |
| 287· | 141·67 | 113·33 | 309· | 153·89 | 123·11 | 332· | 166·67 | 133·33 |
| 287·60 | 142· | 113·60 | 309·20 | 154· | 123·20 | 332·60 | 167· | 133·60 |
| 288· | 142·23 | 113·77 | 310· | 154·45 | 123·55 | 333· | 167·23 | 133·77 |
| 288·50 | 142·50 | 114· | 311· | 155· | 124· | 333·50 | 167·50 | 134· |
| 289· | 142·78 | 114·22 | 312· | 155·56 | 124·44 | 334· | 167·78 | 134·22 |
| 289·40 | 143· | 114·40 | 312·80 | 156· | 124·80 | 334·40 | 168· | 134·40 |
| 290· | 143·34 | 114·66 | 313· | 156·11 | 124·88 | 335· | 168·34 | 134·66 |
| 290·75 | 143·75 | 115· | 313·25 | 156·25 | 125· | 335·75 | 168·75 | 135· |
| 291· | 143·89 | 115·11 | 314· | 156·67 | 125·33 | 336· | 168·89 | 135·11 |

## Table CLXXVIII.—Comparison of Thermometers—*continued.*

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. |
|---|---|---|---|---|---|---|---|---|
| + | + | + | + | + | + | + | + | + |
| 336·20 | 169· | 135·20 | 359·60 | 182· | 145·60 | 381·20 | 194· | 155·20 |
| 337· | 169·45 | 135·55 | 360· | 182·23 | 145·77 | 382· | 194·45 | 155·55 |
| 338· | 170· | 136· | 360·50 | 182·50 | 146· | 383· | 195· | 156· |
| 339· | 170·56 | 136·44 | 361· | 182·78 | 146·22 | 384· | 195·56 | 156·44 |
| 339·80 | 171· | 136·80 | 361·40 | 183· | 146·40 | 384·80 | 196· | 156·80 |
| 340· | 171·11 | 136·88 | 362· | 183·34 | 146·66 | 385· | 196·11 | 156·88 |
| 340·25 | 171·25 | 137· | 362·75 | 183·75 | 147· | 385·25 | 196·25 | 157· |
| 341· | 171·67 | 137·33 | 363· | 183·89 | 147·11 | 386· | 196·67 | 157·33 |
| 341·60 | 172· | 137·60 | 363·20 | 184· | 147·20 | 386·60 | 197· | 157·60 |
| 342· | 172·23 | 137·77 | 364· | 184·45 | 147·55 | 387· | 197·23 | 157·77 |
| 342·50 | 172·50 | 138· | 365· | 185· | 148· | 387·50 | 197·50 | 158· |
| 343· | 172·78 | 138·22 | 366· | 185·56 | 148·44 | 388· | 197·78 | 158·22 |
| 343·40 | 173· | 138·40 | 366·80 | 186· | 148·80 | 388·40 | 198· | 158·40 |
| 344· | 173·34 | 138·66 | 367· | 186·11 | 148·88 | 389· | 198·34 | 158·66 |
| 344·75 | 173·75 | 139· | 367·25 | 186·25 | 149· | 389·75 | 198·75 | 159· |
| 345· | 173·89 | 139·11 | 368· | 186·67 | 149·33 | 390· | 198·89 | 159·11 |
| 345·20 | 174· | 139·20 | 368·60 | 187· | 149·60 | 390·20 | 199· | 159·20 |
| 346· | 174·45 | 139·55 | 369· | 187·23 | 149·77 | 391· | 199·45 | 159·55 |
| 347· | 175· | 140· | 369·50 | 187·50 | 150· | 392· | 200· | 160· |
| 348· | 175·56 | 140·44 | 370· | 187·78 | 150·22 | 393· | 200·56 | 160·44 |
| 348·80 | 176· | 140·80 | 370·40 | 188· | 150·40 | 393·80 | 201· | 160·80 |
| 349· | 176·11 | 140·88 | 371· | 188·34 | 150·66 | 394· | 201·11 | 160·88 |
| 349·25 | 176·25 | 141· | 371·75 | 188·75 | 151· | 394·25 | 201·25 | 161· |
| 350· | 176·67 | 141·33 | 372· | 188·89 | 151·11 | 395· | 201·67 | 161·33 |
| 350·60 | 177· | 141·60 | 372·20 | 189· | 151·20 | 395·60 | 202· | 161·60 |
| 351· | 177·23 | 141·77 | 373· | 189·45 | 151·55 | 396· | 202·23 | 161·77 |
| 351·50 | 177·50 | 142· | 374· | 190· | 152· | 396·50 | 202·50 | 162· |
| 352· | 177·78 | 142·22 | 375· | 190·56 | 152·44 | 397· | 202·78 | 162·22 |
| 352·40 | 178· | 142·40 | 375·80 | 191· | 152·80 | 397·40 | 203· | 162·40 |
| 353· | 178·34 | 142·66 | 376· | 191·11 | 152·88 | 398· | 203·34 | 162·66 |
| 353·75 | 178·75 | 143· | 376·25 | 191·25 | 153· | 398·75 | 203·75 | 163· |
| 354· | 178·89 | 143·11 | 377· | 191·67 | 153·33 | 399· | 203·89 | 163·11 |
| 354·20 | 179· | 143·20 | 377·60 | 192· | 153·60 | 399·20 | 204· | 163·20 |
| 355· | 179·45 | 143·55 | 378· | 192·23 | 153·77 | 400· | 204·45 | 163·55 |
| 356· | 180· | 144· | 378·50 | 192·50 | 154· | 401· | 205· | 164· |
| 357· | 180·56 | 144·44 | 379· | 192·78 | 154·22 | 402· | 205·56 | 164·44 |
| 357·80 | 181· | 144·80 | 379·40 | 193· | 154·40 | 402·80 | 206· | 164·80 |
| 358· | 181·11 | 144·88 | 380· | 193·34 | 154·66 | 403· | 206·11 | 164·88 |
| 358·25 | 181·25 | 145· | 380·75 | 193·75 | 155· | 403·25 | 206·25 | 165· |
| 359· | 181·67 | 145·33 | 381· | 193·89 | 155·11 | 404· | 206·67 | 165·33 |

## Table CLXXVIII.—Comparison of Thermometers—*continued.*

| Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. | Fahrenheit degs. | Centigrade degs. | Reaumur degs. |
|---|---|---|---|---|---|---|---|---|
| + | + | + | + | + | + | + | + | + |
| 404·60 | 207· | 165·60 | 426· | 218·89 | 175·11 | 448·25 | 231·25 | 185· |
| 405· | 207·23 | 165·77 | 426·20 | 219· | 175·20 | 449· | 231·67 | 185·33 |
| 405·50 | 207·50 | 166· | 427· | 219·45 | 175·55 | 449·60 | 232· | 185·60 |
| 406· | 208·75 | 163·22 | 428· | 220· | 176· | 450· | 232·23 | 185·77 |
| 406·40 | 208· | 166·40 | 429· | 220·56 | 176·44 | 455· | 235· | 188· |
| 407· | 208·34 | 166·66 | 429·80 | 221· | 176·80 | 464· | 240· | 192· |
| 407·75 | 208·75 | 167· | 430· | 221·11 | 176·88 | 473· | 245· | 196· |
| 408· | 208·89 | 167·11 | 430·25 | 221·25 | 177· | 482· | 250· | 200· |
| 408·20 | 209· | 167·20 | 431· | 221·67 | 177·33 | 491· | 255· | 204· |
| 409· | 209·45 | 167·55 | 431·60 | 222· | 177·60 | 500· | 260· | 208· |
| 410· | 210· | 168· | 432· | 222·23 | 177·77 | 509· | 265· | 212· |
| 411· | 210·56 | 168·44 | 432·50 | 222·50 | 178· | 518· | 270· | 216· |
| 411·80 | 211· | 168·80 | 433· | 222·78 | 178·22 | 527· | 275· | 220· |
| 412· | 211·11 | 168·88 | 433·40 | 223· | 178·40 | 536· | 280· | 224· |
| 412·25 | 211·25 | 169· | 434· | 223·34 | 178·66 | 545· | 285· | 228· |
| 413· | 211·67 | 169·33 | 434·75 | 223·75 | 179· | 554· | 290· | 232· |
| 413·60 | 212· | 169·60 | 435· | 223·89 | 179·11 | 563· | 295· | 236· |
| 414· | 212·23 | 169·77 | 435·20 | 224· | 179·20 | 572· | 300· | 240· |
| 414·50 | 212·50 | 170· | 436· | 224·45 | 179·55 | 581· | 305· | 244· |
| 415· | 212·78 | 170·22 | 437· | 225· | 180· | 590· | 310· | 248· |
| 415·40 | 213· | 170·40 | 438· | 225·56 | 180·44 | 599· | 315· | 252· |
| 416· | 213·34 | 170·66 | 438·80 | 226· | 180·80 | 608· | 320· | 256· |
| 416·75 | 213·75 | 171· | 439· | 226·11 | 180·88 | 617· | 325· | 260· |
| 417· | 213·89 | 171·11 | 439·25 | 226·25 | 181· | 626· | 330· | 264· |
| 417·20 | 214· | 171·20 | 440· | 226·67 | 181·33 | 635· | 335· | 268· |
| 418· | 214·45 | 171·55 | 440·60 | 227· | 181·60 | 644· | 340· | 272· |
| 419· | 215· | 172· | 441· | 227·23 | 181·77 | 653· | 345· | 276· |
| 420· | 215·56 | 172·44 | 441·50 | 227·50 | 182· | 662· | 350· | 280· |
| 420·80 | 216· | 172·80 | 442· | 227·78 | 182·22 | 671· | 355· | 284· |
| 421· | 216·11 | 172·88 | 442·40 | 228· | 182·40 | 680· | 360· | 288· |
| 421·25 | 216·25 | 173· | 443· | 228·34 | 182·66 | 689· | 365· | 292· |
| 422· | 216·67 | 173·33 | 443·75 | 228·75 | 183· | 698· | 370· | 296· |
| 422·60 | 217· | 173·60 | 444· | 228·89 | 183·11 | 707· | 375· | 300· |
| 423· | 217·23 | 173·77 | 444·20 | 229· | 183·20 | 716· | 380· | 304· |
| 423·50 | 217·50 | 174· | 445· | 229·45 | 183·55 | 725· | 385· | 308· |
| 424· | 217·78 | 174·22 | 446· | 230· | 184· | 734· | 390· | 312· |
| 424·40 | 218· | 174·40 | 447· | 230·56 | 184·44 | 743· | 395· | 316· |
| 425· | 218·34 | 174·66 | 447·80 | 231· | 184·80 | 752· | 400· | 320· |
| 425·75 | 218·75 | 175· | 448· | 231·11 | 184·88 |  |  |  |

## Table CLXXIX.—Properties of Saturated Steam.

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| −14 | 90·4 | 28740 | ·002170 | 460·7 | 1051·1 | 1109·5 | ·7 |
| −13 | 120·3 | 12480 | ·004998 | 200·1 | 1030·3 | 1118·6 | 1·7 |
| −12 | 137·5 | 8080 | ·007720 | 129·5 | 1018·3 | 1123·9 | 2·7 |
| −11 | 149·8 | 6009 | ·01038 | 96·32 | 1009·7 | 1127·6 | 3·7 |
| −10 | 159·7 | 4799 | ·01300 | 76·92 | 1002·8 | 1130·7 | 4·7 |
| −9 | 167·9 | 4003 | ·01558 | 64·17 | 997·0 | 1133·2 | 5·7 |
| −8 | 174·9 | 3439 | ·01814 | 55·12 | 992·1 | 1135·3 | 6·7 |
| −7 | 181·1 | 3010 | ·02072 | 48·25 | 987·8 | 1137·2 | 7·7 |
| −6 | 186·7 | 2690 | ·02319 | 43·12 | 983·9 | 1138·9 | 8·7 |
| −5 | 191·8 | 2428 | ·02568 | 38·93 | 980·3 | 1140·4 | 9·7 |
| −4 | 196·4 | 2215 | ·02817 | 35·50 | 977·1 | 1141·9 | 10·7 |
| −3 | 200·7 | 2036 | ·03063 | 32·64 | 974·0 | 1143·2 | 11·7 |
| −2 | 204·7 | 1886 | ·03307 | 30·24 | 971·2 | 1144·4 | 12·7 |
| −1 | 208·5 | 1755 | ·03553 | 28·14 | 968·5 | 1145·5 | 13·7 |
| Atmosphere. | 212·0 | 1643 | ·03797 | 26·34 | 966·1 | 1146·6 | 14·7 |
| 1 | 215·3 | 1544 | ·04039 | 24·76 | 963·8 | 1147·6 | 15·7 |
| 2 | 218·5 | 1457 | ·04280 | 23·36 | 961·5 | 1148·6 | 16·7 |
| 3 | 221·5 | 1380 | ·04521 | 22·12 | 959·4 | 1149·5 | 17·7 |
| 4 | 224·4 | 1310 | ·04761 | 21·0 | 957·3 | 1150·4 | 18·7 |
| 5 | 227·1 | 1248 | ·05000 | 20·0 | 955·4 | 1151·2 | 19·7 |
| 6 | 229·8 | 1191 | ·05238 | 19·09 | 953·5 | 1152·0 | 20·7 |
| 7 | 232·3 | 1139 | ·05475 | 18·26 | 951·8 | 1152·8 | 21·7 |
| 8 | 234·7 | 1092 | ·05712 | 17·51 | 950·1 | 1153·5 | 22·7 |
| 9 | 237·1 | 1049 | ·05949 | 16·81 | 948·4 | 1154·3 | 23·7 |
| 10 | 239·4 | 1009 | ·06184 | 16·17 | 946·7 | 1155·0 | 24·7 |
| 11 | 241·6 | 971·8 | ·06419 | 15·58 | 945·2 | 1155·6 | 25·7 |
| 12 | 243·7 | 937·5 | ·06654 | 15·03 | 943·7 | 1156·3 | 26·7 |
| 13 | 245·8 | 905·7 | ·06888 | 14·52 | 942·2 | 1156·9 | 27·7 |
| 14 | 247·8 | 876·0 | ·07122 | 14·04 | 940·8 | 1157·5 | 28·7 |
| 15 | 249·7 | 818·2 | ·07355 | 13·60 | 939·4 | 1158·1 | 29·7 |
| 16 | 251·6 | 822·2 | ·07587 | 13·18 | 938·1 | 1158·7 | 30·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 17 | 253·5 | 797·8 | ·07819 | 12·79 | 936·8 | 1159·3 | 31·7 |
| 18 | 255·3 | 774·9 | ·08050 | 12·42 | 935·5 | 1159·8 | 32·7 |
| 19 | 257·0 | 753·2 | ·08282 | 12·07 | 934·3 | 1160·3 | 33·7 |
| 20 | 258·7 | 732·8 | ·08513 | 11·75 | 933·1 | 1160·9 | 34·7 |
| 21 | 260·4 | 713·5 | ·08743 | 11·44 | 931·9 | 1161·4 | 35·7 |
| 22 | 262·0 | 695·2 | ·08973 | 11·14 | 930·7 | 1161·9 | 36·7 |
| 23 | 263·6 | 677·9 | ·09203 | 10·87 | 929·6 | 1162·3 | 37·7 |
| 24 | 265·2 | 661·4 | ·09433 | 10·60 | 928·4 | 1162·8 | 38·7 |
| 25 | 266·7 | 645·7 | ·09661 | 10·35 | 927·4 | 1163·3 | 39·7 |
| 26 | 268·2 | 630·8 | ·09890 | 10·11 | 926·3 | 1163·8 | 40·7 |
| 27 | 269·7 | 616·6 | ·1011 | 9·883 | 925·2 | 1164·2 | 41·7 |
| 28 | 271·1 | 603·0 | ·1034 | 9·666 | 924·3 | 1164·6 | 42·7 |
| 29 | 272·6 | 590·0 | ·1057 | 9·458 | 923·2 | 1165·1 | 43·7 |
| 30 | 273·9 | 577·6 | ·1080 | 9·259 | 922·3 | 1165·5 | 44·7 |
| 31 | 275·3 | 565·7 | ·1102 | 9·068 | 921·3 | 1165·9 | 45·7 |
| 32 | 276·7 | 554·3 | ·1125 | 8·885 | 920·3 | 1166·3 | 46·7 |
| 33 | 278·0 | 543·4 | ·1148 | 8·710 | 919·3 | 1166·7 | 47·7 |
| 34 | 279·3 | 532·9 | ·1170 | 8·541 | 918·4 | 1167·1 | 48·7 |
| 35 | 280·5 | 522·8 | ·1193 | 8·380 | 917·6 | 1167·5 | 49·7 |
| 36 | 281·8 | 513·1 | ·1215 | 8·225 | 916·6 | 1167·9 | 50·7 |
| 37 | 283·0 | 503·8 | ·1238 | 8·075 | 915·8 | 1168·3 | 51·7 |
| 38 | 284·2 | 494·8 | ·1260 | 7·931 | 914·9 | 1168·6 | 52·7 |
| 39 | 285·4 | 486·1 | ·1283 | 7·792 | 914·1 | 1169·0 | 53·7 |
| 40 | 286·6 | 477·7 | ·1305 | 7·658 | 913·2 | 1169·4 | 54·7 |
| 41 | 287·8 | 469·7 | ·1328 | 7·529 | 912·3 | 1169·7 | 55·7 |
| 42 | 288·9 | 461·9 | ·1350 | 7·404 | 911·6 | 1170·1 | 56·7 |
| 43 | 290·1 | 454·4 | ·1373 | 7·283 | 910·7 | 1170·4 | 57·7 |
| 44 | 291·2 | 447·1 | ·1395 | 7·167 | 909·9 | 1170·8 | 58·7 |
| 45 | 292·3 | 440·0 | ·1417 | 7·054 | 909·1 | 1171·1 | 59·7 |
| 46 | 293·4 | 433·2 | ·1440 | 6·944 | 908·3 | 1171·4 | 60·7 |
| 47 | 294·4 | 426·6 | ·1462 | 6·839 | 907·6 | 1171·7 | 61·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 48 | 295·5 | 420·2 | ·1484 | 6·736 | 906·8 | 1172·1 | 62·7 |
| 49 | 296·5 | 414·1 | ·1506 | 6·636 | 906·1 | 1172·4 | 63·7 |
| 50 | 297·5 | 408·0 | ·1529 | 6·540 | 905·4 | 1172·7 | 64·7 |
| 51 | 298·6 | 402·2 | ·1551 | 6·446 | 904·6 | 1173·0 | 65·7 |
| 52 | 299·6 | 396·5 | ·1573 | 6·356 | 903·9 | 1173·3 | 66·7 |
| 53 | 300·6 | 391·0 | ·1595 | 6·267 | 903·2 | 1173·6 | 67·7 |
| 54 | 301·5 | 385·6 | ·1617 | 6·181 | 902·5 | 1173·9 | 68·7 |
| 55 | 302·5 | 380·4 | ·1639 | 6·098 | 901·8 | 1174·2 | 69·7 |
| 56 | 303·5 | 375·4 | ·1661 | 6·017 | 901·1 | 1174·5 | 70·7 |
| 57 | 304·4 | 370·4 | ·1684 | 5·938 | 900·5 | 1174·8 | 71·7 |
| 58 | 305·3 | 365·6 | ·1706 | 5·861 | 899·8 | 1175·1 | 72·7 |
| 59 | 306·3 | 361·0 | ·1728 | 5·786 | 899·1 | 1175·4 | 73·7 |
| 60 | 307·2 | 356·4 | ·1750 | 5·714 | 898·5 | 1175·6 | 74·7 |
| 61 | 308·1 | 352·0 | ·1772 | 5·643 | 897·8 | 1175·9 | 75·7 |
| 62 | 309·0 | 347·7 | ·1794 | 5·573 | 897·2 | 1176·2 | 76·7 |
| 63 | 309·9 | 343·5 | ·1816 | 5·506 | 896·5 | 1176·5 | 77·7 |
| 64 | 310·8 | 339·4 | ·1838 | 5·440 | 895·9 | 1176·7 | 78·7 |
| 65 | 311·6 | 335·4 | ·1860 | 5·376 | 895·3 | 1177·0 | 79·7 |
| 66 | 312·5 | 331·5 | ·1882 | 5·313 | 894·7 | 1177·3 | 80·7 |
| 67 | 313·3 | 327·7 | ·1904 | 5·252 | 894·1 | 1177·5 | 81·7 |
| 68 | 314·2 | 323·9 | ·1926 | 5·192 | 893·4 | 1177·8 | 82·7 |
| 69 | 315·0 | 320·3 | ·1947 | 5·134 | 892·9 | 1178·0 | 83·7 |
| 70 | 315·8 | 316·7 | ·1969 | 5·077 | 892·3 | 1178·3 | 84·7 |
| 71 | 316·7 | 313·3 | ·1991 | 5·021 | 891·6 | 1178·5 | 85·7 |
| 72 | 317·5 | 309·9 | ·2013 | 4·967 | 891·1 | 1178·8 | 86·7 |
| 73 | 318·3 | 306·5 | ·2035 | 4·914 | 890·5 | 1179·0 | 87·7 |
| 74 | 319·1 | 303·3 | ·2056 | 4·862 | 889·9 | 1179·3 | 88·7 |
| 75 | 319·9 | 300·1 | ·2078 | 4·811 | 889·3 | 1179·5 | 89·7 |
| 76 | 320·7 | 297·0 | ·2100 | 4·761 | 888·8 | 1179·8 | 90·7 |
| 77 | 321·4 | 294·0 | ·2122 | 4·712 | 888·3 | 1180·0 | 91·7 |
| 78 | 322·2 | 291·0 | ·2144 | 4·664 | 887·7 | 1180·2 | 92·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 79 | 323·0 | 288·0 | ·2165 | 4·617 | 887·1 | 1180·5 | 93·7 |
| 80 | 323·8 | 285·2 | ·2187 | 4·572 | 886·5 | 1180·7 | 94·7 |
| 81 | 324·5 | 282·4 | ·2209 | 4·527 | 886·0 | 1180·9 | 95·7 |
| 82 | 325·2 | 279·6 | ·2230 | 4·483 | 885·5 | 1181·1 | 96·7 |
| 83 | 326·0 | 276·9 | ·2252 | 4·439 | 885·0 | 1181·4 | 97·7 |
| 84 | 326·7 | 274·3 | ·2274 | 4·397 | 884·4 | 1181·6 | 98·7 |
| 85 | 327·4 | 271·7 | ·2295 | 4·356 | 883·9 | 1181·8 | 99·7 |
| 86 | 328·1 | 269·2 | ·2317 | 4·315 | 883·4 | 1182·0 | 100·7 |
| 87 | 328·9 | 266·7 | ·2339 | 4·275 | 882·9 | 1182·3 | 101·7 |
| 88 | 329·6 | 264·3 | ·2360 | 4·236 | 882·4 | 1182·5 | 102·7 |
| 89 | 330·3 | 261·9 | ·2382 | 4·198 | 881·9 | 1182·7 | 103·7 |
| 90 | 331·0 | 259·5 | ·2404 | 4·160 | 881·3 | 1182·9 | 104·7 |
| 91 | 331·7 | 257·2 | ·2425 | 4·123 | 880·8 | 1183·1 | 105·7 |
| 92 | 332·3 | 254·9 | ·2447 | 4·087 | 880·4 | 1183·3 | 106·7 |
| 93 | 333·0 | 252·5 | ·2468 | 4·051 | 879·9 | 1183·5 | 107·7 |
| 94 | 333·7 | 250·5 | ·2490 | 4·016 | 879·4 | 1183·7 | 108·7 |
| 95 | 334·4 | 248·4 | ·2511 | 3·981 | 878·9 | 1183·9 | 109·7 |
| 96 | 335·1 | 246·3 | ·2533 | 3·948 | 878·4 | 1184·2 | 110·7 |
| 97 | 335·7 | 244·2 | ·2554 | 3·914 | 878·0 | 1184·3 | 111·7 |
| 98 | 336·4 | 242·2 | ·2576 | 3·882 | 877·4 | 1184·6 | 112·7 |
| 99 | 337·0 | 240·1 | ·2598 | 3·849 | 877·0 | 1184·7 | 113·7 |
| 100 | 337·7 | 238·2 | ·2619 | 3·818 | 876·5 | 1184·9 | 114·7 |
| 101 | 338·3 | 236·2 | ·2640 | 3·787 | 876·1 | 1185·1 | 115·7 |
| 102 | 339·0 | 234·3 | ·2662 | 3·757 | 875·6 | 1185·3 | 116·7 |
| 103 | 339·6 | 232·5 | ·2683 | 3·727 | 875·1 | 1185·5 | 117·7 |
| 104 | 340·2 | 230·6 | ·2704 | 3·697 | 874·7 | 1185·7 | 118·7 |
| 105 | 340·9 | 228·8 | ·2726 | 3·668 | 874·2 | 1185·9 | 119·7 |
| 106 | 341·5 | 227·0 | ·2747 | 3·639 | 873·8 | 1186·1 | 120·7 |
| 107 | 342·1 | 225·1 | ·2769 | 3·611 | 873·3 | 1186·3 | 121·7 |
| 108 | 342·7 | 223·6 | ·2790 | 3·584 | 872·9 | 1186·5 | 122·7 |
| 109 | 343·3 | 221·9 | ·2811 | 3·556 | 872·5 | 1186·7 | 123·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32°, Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 110 | 343·9 | 220·1 | ·2833 | 3·530 | 872·0 | 1186·8 | 124·7 |
| 111 | 344·5 | 218·5 | ·2854 | 3·503 | 871·6 | 1187·0 | 125·7 |
| 112 | 345·1 | 216·9 | ·2875 | 3·477 | 871·2 | 1187·2 | 126·7 |
| 113 | 345·7 | 215·3 | ·2897 | 3·452 | 870·7 | 1187·4 | 127·7 |
| 114 | 346·3 | 213·7 | ·2918 | 3·426 | 870·3 | 1187·6 | 128·7 |
| 115 | 346·9 | 212·2 | ·2939 | 3·401 | 869·9 | 1187·8 | 129·7 |
| 116 | 347·5 | 210·7 | ·2961 | 3·377 | 869·4 | 1187·9 | 130·7 |
| 117 | 348·1 | 209·2 | ·2982 | 3·353 | 869·0 | 1188·1 | 131·7 |
| 118 | 348·7 | 207·7 | ·3003 | 3·329 | 868·6 | 1188·3 | 132·7 |
| 119 | 349·2 | 206·2 | ·3025 | 3·306 | 868·2 | 1188·5 | 133·7 |
| 120 | 349·8 | 204·8 | ·3046 | 3·283 | 867·8 | 1188·6 | 134·7 |
| 121 | 350·4 | 203·4 | ·3067 | 3·260 | 867·3 | 1188·8 | 135·7 |
| 122 | 351·0 | 202·0 | ·3088 | 3·237 | 866·9 | 1189·0 | 136·7 |
| 123 | 351·5 | 200·6 | ·3110 | 3·215 | 866·5 | 1189·2 | 137·7 |
| 124 | 352·1 | 199·2 | ·3131 | 3·194 | 866·1 | 1189·3 | 138·7 |
| 125 | 352·6 | 197·9 | ·3152 | 3·172 | 865·7 | 1189·5 | 139·7 |
| 126 | 353·2 | 196·6 | ·3173 | 3·151 | 865·3 | 1189·7 | 140·7 |
| 127 | 353·7 | 195·3 | ·3194 | 3·130 | 864·9 | 1189·8 | 141·7 |
| 128 | 354·3 | 194·0 | ·3216 | 3·109 | 864·5 | 1190·0 | 142·7 |
| 129 | 354·8 | 192·7 | ·3237 | 3·089 | 864·1 | 1190·2 | 143·7 |
| 130 | 355·4 | 191·5 | ·3258 | 3·069 | 863·7 | 1190·3 | 144·7 |
| 131 | 355·9 | 190·2 | ·3279 | 3·049 | 863·3 | 1190·5 | 145·7 |
| 132 | 356·4 | 189·0 | ·3300 | 3·030 | 863·0 | 1190·7 | 146·7 |
| 133 | 357·0 | 187·8 | ·3321 | 3·010 | 862·5 | 1190·8 | 147·7 |
| 134 | 357·5 | 186·6 | ·3343 | 2·991 | 862·2 | 1191·0 | 148·7 |
| 135 | 358·0 | 185·4 | ·3364 | 2·973 | 861·8 | 1191·1 | 149·7 |
| 136 | 358·5 | 184·3 | ·3385 | 2·954 | 861·4 | 1191·3 | 150·7 |
| 137 | 359·1 | 182·9 | ·3406 | 2·936 | 861·0 | 1191·5 | 151·7 |
| 138 | 359·6 | 182·0 | ·3427 | 2·918 | 860·6 | 1191·6 | 152·7 |
| 139 | 360·1 | 180·9 | ·3448 | 2·900 | 860·3 | 1191·8 | 153·7 |
| 140 | 360·6 | 179·8 | ·3469 | 2·882 | 859·9 | 1191·9 | 154·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 141 | 361·1 | 178·7 | ·3490 | 2·865 | 859·6 | 1192·1 | 155·7 |
| 142 | 361·6 | 177·6 | ·3511 | 2·848 | 859·2 | 1192·2 | 156·7 |
| 143 | 362·1 | 176·6 | ·3532 | 2·831 | 858·8 | 1192·4 | 157·7 |
| 144 | 362·6 | 175·5 | ·3554 | 2·814 | 858·5 | 1192·5 | 158·7 |
| 145 | 363·1 | 174·5 | ·3575 | 2·797 | 858·1 | 1192·7 | 159·7 |
| 146 | 363·6 | 173·5 | ·3596 | 2·781 | 857·7 | 1192·8 | 160·7 |
| 147 | 364·1 | 172·5 | ·3617 | 2·765 | 857·4 | 1193·0 | 161·7 |
| 148 | 364·6 | 171·5 | ·3638 | 2·749 | 857·0 | 1193·2 | 162·7 |
| 149 | 365·1 | 170·5 | ·3659 | 2·733 | 856·6 | 1193·3 | 163·7 |
| 150 | 365·6 | 169·5 | ·3680 | 2·717 | 856·3 | 1193·5 | 164·7 |
| 151 | 366·1 | 168·6 | ·3701 | 2·702 | 855·9 | 1193·6 | 165·7 |
| 152 | 366·6 | 167·6 | ·3722 | 2·687 | 855·5 | 1193·8 | 166·7 |
| 153 | 367·1 | 166·7 | ·3743 | 2·672 | 855·2 | 1193·9 | 167·7 |
| 154 | 367·5 | 165·7 | ·3764 | 2·657 | 854·9 | 1194·0 | 168·7 |
| 155 | 368·0 | 164·8 | ·3785 | 2·642 | 854·5 | 1194·2 | 169·7 |
| 156 | 368·5 | 163·9 | ·3806 | 2·627 | 854·2 | 1194·3 | 170·7 |
| 157 | 369·0 | 163·0 | ·3827 | 2·613 | 853·8 | 1194·5 | 171·7 |
| 158 | 369·4 | 162·1 | ·3847 | 2·599 | 853·5 | 1194·6 | 172·7 |
| 159 | 369·9 | 161·2 | ·3868 | 2·585 | 853·1 | 1194·8 | 173·7 |
| 160 | 370·4 | 160·4 | ·3889 | 2·571 | 852·8 | 1194·9 | 174·7 |
| 161 | 370·8 | 159·5 | ·3910 | 2·557 | 852·5 | 1195·0 | 175·7 |
| 162 | 371·3 | 158·7 | ·3931 | 2·543 | 852·1 | 1195·2 | 176·7 |
| 163 | 371·7 | 157·8 | ·3952 | 2·530 | 851·8 | 1195·3 | 177·7 |
| 164 | 372·2 | 157·0 | ·3973 | 2·517 | 851·5 | 1195·5 | 178·7 |
| 165 | 372·7 | 156·2 | ·3994 | 2·504 | 851·1 | 1195·6 | 179·7 |
| 166 | 373·1 | 155·4 | ·4015 | 2·491 | 850·8 | 1195·7 | 180·7 |
| 167 | 373·6 | 154·6 | ·4036 | 2·478 | 850·4 | 1195·9 | 181·7 |
| 168 | 374·0 | 153·8 | ·4057 | 2·465 | 850·2 | 1196·0 | 182·7 |
| 169 | 374·5 | 153·0 | ·4077 | 2·452 | 849·8 | 1196·2 | 183·7 |
| 170 | 374·9 | 152·2 | ·4098 | 2·440 | 849·5 | 1196·3 | 184·7 |
| 171 | 375·4 | 151·4 | ·4119 | 2·427 | 849·1 | 1196·4 | 185·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square Inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 172 | 375·8 | 150·7 | ·4140 | 2·415 | 848·8 | 1196·6 | 186·7 |
| 173 | 376·2 | 149·9 | ·4161 | 2·403 | 848·5 | 1196·7 | 187·7 |
| 174 | 376·7 | 149·2 | ·4182 | 2·391 | 848·2 | 1196·8 | 188·7 |
| 175 | 377·1 | 148·4 | ·4203 | 2·379 | 847·9 | 1197·0 | 189·7 |
| 176 | 377·5 | 147·7 | ·4223 | 2·368 | 847·6 | 1197·1 | 190·7 |
| 177 | 378·0 | 147·0 | ·4244 | 2·356 | 847·2 | 1197·2 | 191·7 |
| 178 | 378·4 | 146·3 | ·4265 | 2·344 | 846·9 | 1197·4 | 192·7 |
| 179 | 378·8 | 145·5 | ·4286 | 2·333 | 846·7 | 1197·5 | 193·7 |
| 180 | 379·3 | 144·8 | ·4307 | 2·322 | 846·3 | 1197·6 | 194·7 |
| 181 | 379·7 | 144·1 | ·4327 | 2·311 | 846·0 | 1197·8 | 195·7 |
| 182 | 380·1 | 143·5 | ·4348 | 2·300 | 845·7 | 1197·9 | 196·7 |
| 183 | 380·5 | 142·8 | ·4369 | 2·289 | 845·4 | 1198·0 | 197·7 |
| 184 | 381·0 | 142·1 | ·4390 | 2·278 | 845·0 | 1198·2 | 198·7 |
| 185 | 381·4 | 141·4 | ·4410 | 2·267 | 844·8 | 1198·3 | 199·7 |
| 186 | 381·8 | 140·8 | ·4431 | 2·257 | 844·5 | 1198·4 | 200·7 |
| 187 | 382·2 | 140·1 | ·4452 | 2·246 | 844·2 | 1198·5 | 201·7 |
| 188 | 382·6 | 139·5 | ·4473 | 2·236 | 843·9 | 1198·6 | 202·7 |
| 189 | 383·0 | 138·8 | ·4493 | 2·225 | 843·6 | 1198·8 | 203·7 |
| 190 | 383·5 | 138·2 | ·4514 | 2·215 | 843·2 | 1198·9 | 204·7 |
| 191 | 383·9 | 137·5 | ·4535 | 2·205 | 842·9 | 1199·0 | 205·7 |
| 192 | 384·3 | 136·9 | ·4556 | 2·195 | 842·6 | 1199·2 | 206·7 |
| 193 | 384·7 | 136·3 | ·4576 | 2·185 | 842·3 | 1199·3 | 207·7 |
| 194 | 385·1 | 135·7 | ·4597 | 2·175 | 842·0 | 1199·4 | 208·7 |
| 195 | 385·5 | 135·1 | ·4618 | 2·165 | 841·8 | 1199·5 | 209·7 |
| 196 | 385·9 | 134·5 | ·4639 | 2·156 | 841·5 | 1199·6 | 210·7 |
| 197 | 386·3 | 133·9 | ·4659 | 2·146 | 841·2 | 1199·8 | 211·7 |
| 198 | 386·7 | 133·3 | ·4680 | 2·137 | 840·9 | 1199·9 | 212·7 |
| 199 | 387·1 | 132·7 | ·4701 | 2·127 | 840·6 | 1200·0 | 213·7 |
| 200 | 387·5 | 132·1 | ·4721 | 2·118 | 840·3 | 1200·1 | 214·7 |

## Table CLXXIX.—Properties of Saturated Steam—*continued.*

| Pressure per Square inch from Mean Atmospheric Pressure. lbs. | Temperature in Fahrenheit Degrees. | Specific or Relative Volume of the Steam. | Density or Weight of 1 Cubic Foot of the Steam. lb. | Cubic Feet of the Steam per lb. | Latent Heat of Evaporation in Thermal Units per lb. of the Steam. | Total Heat in Thermal Units from 32° Fahrenheit per lb. of the Steam. | Absolute Pressure per Square Inch. lbs. |
|---|---|---|---|---|---|---|---|
| 205 | 389·4 | 129·3 | ·4824 | 2·073 | 838·9 | 1200·7 | 219·7 |
| 210 | 391·3 | 126·5 | ·4926 | 2·030 | 837·5 | 1201·3 | 224·7 |
| 215 | 393·2 | 123·9 | ·5027 | 1·989 | 836·2 | 1201·9 | 229·7 |
| 220 | 395·1 | 121·5 | ·5128 | 1·950 | 834·8 | 1202·5 | 234·7 |
| 225 | 397·0 | 119·2 | ·5230 | 1·912 | 833·4 | 1203·1 | 239·7 |
| 230 | 398·9 | 117·0 | ·5332 | 1·876 | 832·1 | 1203·7 | 244·7 |
| 235 | 400·6 | 114·8 | ·5435 | 1·840 | 830·8 | 1204·2 | 249·7 |
| 240 | 402·3 | 112·6 | ·5538 | 1·805 | 829·5 | 1204·7 | 254·7 |
| 245 | 404·0 | 110·5 | ·5641 | 1·772 | 828·2 | 1205·2 | 259·7 |
| 250 | 405·7 | 108·6 | ·5744 | 1·741 | 826·9 | 1205·8 | 264·7 |
| 255 | 407·4 | 106·7 | ·5846 | 1·711 | 825·7 | 1206·3 | 269·7 |
| 260 | 409·0 | 104·8 | ·5949 | 1·681 | 824·5 | 1206·8 | 274·7 |
| 265 | 410·7 | 103·0 | ·6052 | 1·652 | 823·3 | 1207·3 | 279·7 |
| 270 | 412·3 | 101·3 | ·6155 | 1·624 | 822·1 | 1207·8 | 284·7 |
| 275 | 413·9 | 99·7 | ·6258 | 1·597 | 820·9 | 1208·3 | 289·7 |
| 280 | 415·5 | 98·2 | ·6361 | 1·572 | 819·7 | 1208·8 | 294·7 |
| 285 | 417·0 | 96·7 | ·6462 | 1·548 | 818·6 | 1209·2 | 299·7 |
| 290 | 418·5 | 95·2 | ·6563 | 1·524 | 817·5 | 1209·7 | 304·7 |
| 295 | 420·0 | 93·7 | ·6664 | 1·501 | 816·4 | 1210·2 | 309·7 |
| 300 | 421·5 | 92·3 | ·6766 | 1·478 | 815·3 | 1210·6 | 314·7 |
| 305 | 423·0 | 90·9 | ·6868 | 1·456 | 814·2 | 1211·0 | 319·7 |
| 310 | 424·4 | 89·6 | ·6968 | 1·435 | 813·1 | 1211·5 | 324·7 |
| 315 | 425·8 | 88·3 | ·7068 | 1·415 | 812·1 | 1211·9 | 329·7 |
| 320 | 427·2 | 87·0 | ·7168 | 1·395 | 811·1 | 1212·3 | 334·7 |
| 325 | 428·6 | 85·8 | ·7268 | 1·376 | 810·1 | 1212·7 | 339·7 |
| 350 | 435·4 | 80·3 | ·7770 | 1·287 | 805·0 | 1214·8 | 364·7 |
| 400 | 447·9 | 71·2 | ·8764 | 1·141 | 795·7 | 1218·6 | 414·7 |
| 450 | 459·3 | 63·9 | ·9756 | 1·025 | 787·2 | 1222·1 | 464·7 |
| 500 | 469·9 | 58·1 | 1·0741 | ·931 | 779·2 | 1225·3 | 514·7 |

## Table CLXXIXa.—Total Heat of Evaporation from and at Differing Temperatures.

| From | At 328° Press. 100 | At 335° Press. 110 | At 341° Press. 120 | At 347° Press. 130. | At 353° Press. 140 | At 358° Press. 150. | At 363° Press. 160 | At 368° Press. 170 | At 373° Press. 180 | At 377° Press. 190 | At 381° Press. 200 | At 386° Press. 210 | At 389° Press. 220 | At 393° Press. 230 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 40° F. | 1174 | 1176 | 1178 | 1180 | 1182 | 1183 | 1185 | 1186 | 1188 | 1189 | 1191 | 1192 | 1193 | 1194 |
| 50° | 1164 | 1166 | 1168 | 1170 | 1172 | 1173 | 1175 | 1176 | 1178 | 1179 | 1181 | 1182 | 1183 | 1184 |
| 60° | 1154 | 1156 | 1158 | 1160 | 1162 | 1163 | 1165 | 1166 | 1168 | 1169 | 1171 | 1172 | 1173 | 1174 |
| 70° | 1144 | 1146 | 1148 | 1150 | 1152 | 1153 | 1155 | 1156 | 1158 | 1159 | 1161 | 1162 | 1163 | 1164 |
| 80° | 1134 | 1136 | 1138 | 1140 | 1142 | 1143 | 1145 | 1146 | 1148 | 1149 | 1151 | 1152 | 1153 | 1154 |
| 90° | 1124 | 1126 | 1128 | 1130 | 1132 | 1133 | 1135 | 1136 | 1138 | 1139 | 1141 | 1142 | 1143 | 1144 |
| 100° | 1114 | 1116 | 1118 | 1120 | 1122 | 1123 | 1125 | 1126 | 1128 | 1129 | 1131 | 1132 | 1133 | 1134 |
| 110° | 1104 | 1106 | 1108 | 1110 | 1112 | 1113 | 1115 | 1116 | 1118 | 1119 | 1121 | 1122 | 1123 | 1124 |
| 120° | 1094 | 1096 | 1098 | 1100 | 1102 | 1103 | 1105 | 1106 | 1108 | 1109 | 1111 | 1112 | 1113 | 1114 |
| 130° | 1084 | 1086 | 1088 | 1090 | 1092 | 1093 | 1095 | 1096 | 1098 | 1099 | 1101 | 1102 | 1103 | 1104 |
| 140° | 1074 | 1076 | 1078 | 1080 | 1082 | 1083 | 1085 | 1086 | 1088 | 1089 | 1091 | 1092 | 1093 | 1094 |
| 150° | 1064 | 1066 | 1068 | 1070 | 1072 | 1073 | 1075 | 1076 | 1078 | 1079 | 1081 | 1082 | 1083 | 1084 |
| 160° | 1054 | 1056 | 1058 | 1060 | 1062 | 1063 | 1065 | 1066 | 1068 | 1069 | 1071 | 1072 | 1073 | 1074 |
| 170° | 1044 | 1046 | 1048 | 1050 | 1052 | 1053 | 1055 | 1056 | 1058 | 1059 | 1061 | 1062 | 1063 | 1064 |
| 180° | 1034 | 1036 | 1038 | 1040 | 1042 | 1043 | 1045 | 1046 | 1048 | 1049 | 1051 | 1052 | 1053 | 1054 |
| 190° | 1024 | 1026 | 1028 | 1030 | 1032 | 1033 | 1035 | 1036 | 1038 | 1039 | 1041 | 1042 | 1043 | 1044 |
| 200° | 1014 | 1016 | 1018 | 1020 | 1022 | 1023 | 1025 | 1026 | 1028 | 1029 | 1031 | 1032 | 1033 | 1034 |
| 210° | 1004 | 1006 | 1008 | 1010 | 1012 | 1013 | 1015 | 1016 | 1018 | 1019 | 1021 | 1022 | 1023 | 1024 |
| 212° | 1002 | 1004 | 1006 | 1008 | 1010 | 1011 | 1013 | 1014 | 1016 | 1017 | 1019 | 1020 | 1021 | 1022 |
| 220° | 994 | 996 | 998 | 1000 | 1002 | 1003 | 1005 | 1006 | 1008 | 1009 | 1011 | 1012 | 1013 | 1014 |
| 230° | 984 | 986 | 988 | 990 | 992 | 993 | 995 | 996 | 998 | 999 | 1001 | 1002 | 1003 | 1004 |
| 240° | 974 | 976 | 978 | 980 | 982 | 983 | 985 | 986 | 988 | 989 | 991 | 992 | 993 | 994 |
| 250° | 964 | 966 | 968 | 970 | 972 | 973 | 975 | 976 | 978 | 979 | 981 | 982 | 983 | 984 |

*N.B.*—Pressures are in lbs. absolute; temperatures degrees Fahrenheit.

# ENGLISH AND METRICAL MEASURES.

The English standard yard is on a bronze bar, measured at a temperature of 62°F.

The French standard metre is the length of a platinum bar when at a temperature of 0° Cent. (32°F.).

The Standards Commission, in their Report of 1871–72, considered that a correction was needed to allow for this difference, but the original equivalents were adopted without correction in the Weights and Measures Act of 1878.

| Comparison of | Metre. | Litre. | Kilogramme. |
|---|---|---|---|
| | Inches.* | Gallons.† | Lbs. |
| Corrected (Standards Com.), . | 39·38202 | ·22018 | 2·20462 |
| Adopted (Act of 1878), . . | 39·37079 | ·2200967 | 2·20462 |

### SQUARE MEASURE.

| ins. | feet. | yards. | perches. | roods. | acre. | square metres. |
|---|---|---|---|---|---|---|
| 1 | ·00694 | ·000772 | ·0000255 | ·00000064 | ·000000159 | ·000645 |
| 144 | 1 | ·111 | ·00367 | ·0000918 | ·000023 | ·0929 |
| 1296 | 9 | 1 | ·0331 | ·000826 | ·0002062 | ·8361 |
| 39204 | 272¼ | 30¼ | 1 | ·025 | ·00625 | 25·292 |
| 1568160 | 10890 | 1210 | 40 | 1 | ·25 | 1011·7 |
| 6272640 | 43560 | 4840 | 160 | 4 | 1 | 4046·7 |

10 square chains = 1 acre.
1 hectare = 2·471143 acres.
1 square mile = 27878400 sq. feet.
= 3097600 sq. yards.
= 640 acres.

### CUBIC MEASURE.

| ins. | feet. | yard. | cubic metre or stere. |
|---|---|---|---|
| 1 | ·0005788 | ·000002144 | ·000016386 |
| 1728 | 1 | ·03704 | ·028315 |
| 46656 | 27 | 1 | ·764513 |

### MEASURE OF CAPACITY.

| pints. | gall. | peck. | bushel. | quarter. | wey. | last. | cub. ft. | litres. |
|---|---|---|---|---|---|---|---|---|
| 1 | ·125 | ·0625 | ·01562 | ·00195 | ·00039 | ·000195 | ·020051 | ·5679 |
| 8 | 1 | ·5 | ·125 | ·0156 | ·00312 | ·00156 | ·16046 | 4·543 |
| 16 | 2 | 1 | ·25 | ·03125 | ·00625 | ·00312 | ·32092 | 9·087 |
| 64 | 8 | 4 | 1 | ·125 | ·025 | ·0125 | 1·28367 | 36·34766 |
| 512 | 64 | 32 | 8 | 1 | ·2 | ·1 | 10·269 | 290·781 |
| 2560 | 320 | 160 | 40 | 5 | 1 | ·5 | 51·347 | 1453·906 |
| 5120 | 640 | 320 | 80 | 10 | 2 | 1 | 102·69 | 2907·81 |

1 gallon in wine, ale, or dry measure
= 277·27384 cubic inches = ·16 cubic foot.
= 10 lbs. of distilled water =
Cube feet × 6·2355 = gallons × 10 = lbs.
Cube ins. × ·003607 = gallons.
1 bushel = 2218·19 cubic inches = 1·28 cubic foot.
Cubic feet × ·78 = bushels.

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* At equal temperatures in ordinary air.

† At equal temperatures,—distilled water.

## Table CLXXX.—Nautical Miles (Knots),* Miles, Kilometres, &c.

| Speeds per hour. | | | Feet per minute. | Feet per second. | Speeds per hour. | | | Feet per minute. | Feet per second. |
|---|---|---|---|---|---|---|---|---|---|
| Admiralty Knots. | Statute Miles. | Kilometres. | | | Admiralty Knots. | Statute Miles. | Kilometres. | | |
| 4 | 4·6061 | 7·413 | 405·3 | 6·75 | 10 | 11·5151 | 18·532 | 1013·3 | 16·89 |
| 4·25 | 4·8939 | 7·876 | 430·7 | 7·17 | 10·25 | 11·8030 | 18·995 | 1038·7 | 17·31 |
| 4·5 | 5·1818 | 8·339 | 456 | 7·59 | 10·5 | 12·0909 | 19·458 | 1064 | 17·73 |
| 4·75 | 5·4697 | 8·802 | 481·3 | 8·02 | 10·75 | 12·3788 | 19·921 | 1089·3 | 18·15 |
| 5 | 5·7576 | 9·266 | 506·7 | 8·44 | 11 | 12·6667 | 20·385 | 1114·7 | 18·58 |
| 5·25 | 6·0454 | 9·729 | 532 | 8·86 | 11·25 | 12·9545 | 20·848 | 1140 | 19·00 |
| 5·5 | 6·3333 | 10·192 | 557·3 | 9·28 | 11·5 | 13·2424 | 21·311 | 1165·3 | 19·42 |
| 5·75 | 6·6212 | 10·656 | 582·7 | 9·71 | 11·75 | 13·5303 | 21·775 | 1190·7 | 19·85 |
| 6 | 6·9091 | 11·119 | 608 | 10·13 | 12 | 13·8182 | 22·238 | 1216 | 20·27 |
| 6·25 | 7·1970 | 11·582 | 633·3 | 10·55 | 12·25 | 14·1061 | 22·701 | 1241·3 | 20·69 |
| 6·5 | 7·4848 | 12·046 | 658·7 | 10·97 | 12·5 | 14·3939 | 23·165 | 1266·7 | 21·11 |
| 6·75 | 7·7727 | 12·509 | 684 | 11·40 | 12·75 | 14·6818 | 23·628 | 1292 | 21·54 |
| 7 | 8·0606 | 12·927 | 709·3 | 11·82 | 13 | 14·9697 | 24·091 | 1317·3 | 21·95 |
| 7·25 | 8·3485 | 13·435 | 734·7 | 12·24 | 13·25 | 15·2576 | 24·554 | 1342·7 | 22·38 |
| 7·5 | 8·6364 | 13·899 | 760 | 12·66 | 13·5 | 15·5454 | 25·018 | 1368 | 22·80 |
| 7·75 | 8·9242 | 14·362 | 785·3 | 13·09 | 13·75 | 15·8333 | 25·481 | 1393·3 | 23·22 |
| 8 | 9·2121 | 14·825 | 810·7 | 13·51 | 14 | 16·1212 | 25·944 | 1418·7 | 23·64 |
| 8·25 | 9·5000 | 15·289 | 836 | 13·93 | 14·25 | 16·4091 | 26·408 | 1444 | 24·07 |
| 8·5 | 9·7879 | 15·752 | 861·3 | 14·35 | 14·5 | 16·6970 | 26·871 | 1469·3 | 24·49 |
| 8·75 | 10·0757 | 16·215 | 886·7 | 14·78 | 14·75 | 16·9848 | 27·334 | 1494·7 | 24·91 |
| 9 | 10·3636 | 16·678 | 912 | 15·20 | 15 | 17·2727 | 27·797 | 1520 | 25·33 |
| 9·25 | 10·6515 | 17·142 | 937·3 | 15·62 | 15·25 | 17·5606 | 28·261 | 1545·3 | 25·76 |
| 9·5 | 10·9394 | 17·605 | 962·7 | 16·04 | 15·5 | 17·8485 | 28·724 | 1570·7 | 26·18 |
| 9·75 | 11·2273 | 18·068 | 988 | 16·47 | 15·75 | 18·1364 | 29·187 | 1596 | 26·60 |

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* *N.B.*—Strictly speaking, a knot is a rate of speed and not a measure of distance

## Table CLXXX.—Nautical Miles (Knots), Miles, Kilometres, &c.—*continued.*

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| 16 | 18·4242 | 29·651 | 1621·3 | 27·02 | 23·25 | 26·7727 | 43·086 | 2356 | 39·27 |
| 16·25 | 18·7121 | 30·114 | 1646·7 | 27·45 | 23·5 | 27·0606 | 43·549 | 2381·3 | 39·69 |
| 16·5 | 19·0000 | 30·577 | 1672 | 27·87 | 23·75 | 27·3485 | 44·013 | 2406·7 | 40·11 |
| 16·75 | 19·2879 | 31·041 | 1697·3 | 28·29 | 24 | 27·6364 | 44·476 | 2432 | 40·53 |
| 17 | 19·5757 | 31·504 | 1722·7 | 28·71 | 24·25 | 27·9242 | 44·939 | 2457·3 | 40·95 |
| 17·25 | 19·8636 | 31·967 | 1748 | 29·13 | 24·5 | 28·2121 | 45·403 | 2482·7 | 41·38 |
| 17·5 | 20·1515 | 32·430 | 1773·3 | 29·55 | 24·75 | 28·5000 | 45·866 | 2508 | 41·80 |
| 17·75 | 20·4394 | 32·894 | 1798·7 | 29·98 | 25 | 28·7879 | 46·329 | 2533·3 | 42·22 |
| 18 | 20·7273 | 33·357 | 1824 | 30·40 | 25·25 | 29·0757 | 46·792 | 2558·7 | 42·64 |
| 18·25 | 21·0151 | 33·820 | 1849·3 | 30·82 | 25·5 | 29·3636 | 47·256 | 2584 | 43·07 |
| 18·5 | 21·3030 | 34·284 | 1874·7 | 31·24 | 25·75 | 29·6515 | 47·719 | 2609·3 | 43·49 |
| 18·75 | 21·5909 | 34·747 | 1900 | 31·67 | 26 | 29·9394 | 48·182 | 2634·7 | 43·91 |
| 19 | 21·8788 | 35·210 | 1925·3 | 32·09 | 26·25 | 30·2273 | 48·646 | 2660 | 44·33 |
| 19·25 | 22·1667 | 35·673 | 1950·7 | 32·51 | 26·5 | 30·5151 | 49·109 | 2685·3 | 44·75 |
| 19·5 | 22·4545 | 36·137 | 1976 | 32·93 | 26·75 | 30·8030 | 49·572 | 2710·7 | 45·18 |
| 19·75 | 22·7424 | 36·600 | 2001·3 | 33·35 | 27 | 31·0909 | 50·035 | 2736 | 45·60 |
| 20 | 23·0303 | 37·063 | 2026·7 | 33·78 | 27·25 | 31·3788 | 50·499 | 2761·3 | 46·02 |
| 20·25 | 23·3182 | 37·526 | 2052 | 34·20 | 27·5 | 31·6667 | 50·962 | 2786·7 | 46·44 |
| 20·5 | 23·6061 | 37·990 | 2077·3 | 34·62 | 27·75 | 31·9545 | 51·425 | 2812 | 46·87 |
| 20·75 | 23·8939 | 38·453 | 2102·7 | 35·04 | 28 | 32·2424 | 51·889 | 2837·3 | 47·29 |
| 21 | 24·1818 | 38·916 | 2128 | 35·47 | 28·25 | 32·5303 | 52·352 | 2862·7 | 47·71 |
| 21·25 | 24·4697 | 39·379 | 2153·3 | 35·89 | 28·5 | 32·8182 | 52·815 | 2888 | 48·13 |
| 21·5 | 24·7576 | 39·843 | 2178·7 | 36·31 | 28·75 | 33·1061 | 53·279 | 2913·3 | 48·55 |
| 21·75 | 25·0454 | 40·306 | 2204 | 36·73 | 29 | 33·3939 | 53·742 | 2938·7 | 48·98 |
| 22 | 25·3333 | 40·769 | 2229·3 | 37·15 | 29·25 | 33·6818 | 54·205 | 2964 | 49·40 |
| 22·25 | 25·6212 | 41·233 | 2254·7 | 37·58 | 29·5 | 33·9697 | 54·668 | 2989·3 | 49·82 |
| 22·5 | 25·9091 | 41·696 | 2280 | 38·00 | 29·75 | 34·2576 | 55·132 | 3014·7 | 50·24 |
| 22·75 | 26·1970 | 42·160 | 2305·3 | 38·42 | 30 | 34·5454 | 55·595 | 3040 | 50·67 |
| 23 | 26·4848 | 42·623 | 2330·7 | 38·84 | 30·25 | 34·8333 | 56·058 | 3065·3 | 51·09 |

## Table CLXXX.—Nautical Miles (Knots), Miles, Kilometres, &c.—*continued.*

| Speeds per hour. | | | Feet per minute. | Feet per second. | Speeds per hour. | | | Feet per minute. | Feet per second. |
|---|---|---|---|---|---|---|---|---|---|
| Admiralty Knots. | Statute Miles. | Kilometres. | | | Admiralty Knots. | Statute Miles. | Kilometres. | | |
| 30·5 | 35·1212 | 56·522 | 3090·7 | 51·52 | 34·25 | 39·4393 | 63·471 | 3470·7 | 57·84 |
| 30·75 | 35·4091 | 56·985 | 3116 | 51·94 | 34·5 | 39·7272 | 63·934 | 3496 | 58·26 |
| 31 | 35·6970 | 57·448 | 3141·3 | 52·36 | 34·75 | 40·0151 | 64·397 | 3521·3 | 58·68 |
| 31·25 | 35·9849 | 57·911 | 3166·7 | 52·78 | 35 | 40·3030 | 64·860 | 3546·7 | 59·10 |
| 31·5 | 36·2728 | 58·374 | 3192 | 53·20 | 35·25 | 40·5909 | 65·324 | 3572 | 59·53 |
| 31·75 | 36·5606 | 58·838 | 3217·3 | 53·62 | 35·5 | 40·8788 | 65·788 | 3597·3 | 59·96 |
| 32 | 36·8484 | 59·302 | 3242·7 | 54·04 | 35·75 | 41·1667 | 66·251 | 3622·7 | 60·38 |
| 32·25 | 37·1363 | 59·765 | 3268 | 54·47 | 36 | 41·4546 | 66·714 | 3648 | 60·80 |
| 32·5 | 37·4242 | 60·228 | 3293·3 | 54·90 | 36·25 | 41·7424 | 67·177 | 3673·3 | 61·22 |
| 32·75 | 37·7121 | 60·691 | 3318·7 | 55·32 | 36·5 | 42·0302 | 67·640 | 3698·7 | 61·64 |
| 33 | 38·0000 | 61·154 | 3344 | 55·74 | 36·75 | 42·3181 | 68·104 | 3724 | 62·06 |
| 33·25 | 38·2879 | 61·618 | 3369·3 | 56·16 | 37 | 42·6060 | 68·568 | 3749·3 | 62·48 |
| 33·5 | 38·5758 | 62·082 | 3394·7 | 56·58 | 37·25 | 42·8939 | 69·031 | 3774·7 | 62·91 |
| 33·75 | 38·8636 | 62·545 | 3420 | 57·00 | 37·5 | 43·1818 | 69·494 | 3800 | 63·34 |
| 34 | 39·1514 | 63·008 | 3445·3 | 57·42 | 37·75 | 43·4697 | 69·957 | 3825·3 | 63·77 |

| | Statute Mile. | Kilometre. | Feet per min. | Foot per sec. |
|---|---|---|---|---|
| $\frac{1}{10}$ Knot = | ·11515 | ·1853 | 10·133 | ·1688 |
| $\frac{1}{100}$ „ = | ·011515 | ·01853 | 1·0133 | ·01688 |

## Table CLXXXI.—Kilometres and Admiralty Knots.

| Kilometres. | Admiralty Knots. | Kilometres. | Admiralty Knots. | Kilometres. | Admiralty Knots. | Kilometres. | Admiralty Knots. |
|---|---|---|---|---|---|---|---|
| 7·5 | 4·047 | 23·5 | 12·681 | 39·5 | 21·315 | 55·5 | 29·949 |
| 8 | 4·317 | 24 | 12·951 | 40 | 21·585 | 56 | 30·218 |
| 8·5 | 4·587 | 24·5 | 13·221 | 40·5 | 21·854 | 56·5 | 30·488 |
| 9 | 4·857 | 25 | 13·490 | 41 | 22·124 | 57 | 30·758 |
| 9·5 | 5·126 | 25·5 | 13·760 | 41·5 | 22·394 | 57·5 | 31·028 |
| 10 | 5·396 | 26 | 14·030 | 42 | 22·664 | 58 | 31·298 |
| 10·5 | 5·666 | 26·5 | 14·300 | 42·5 | 22·934 | 58·5 | 31·568 |
| 11 | 5·936 | 27 | 14·570 | 43 | 23·203 | 59 | 31·838 |
| 11·5 | 6·206 | 27·5 | 14·839 | 43·5 | 23·473 | 59·5 | 32·107 |
| 12 | 6·475 | 28 | 15·109 | 44 | 23·743 | 60 | 32·376 |
| 12·5 | 6·745 | 28·5 | 15·379 | 44·5 | 24·013 | 60·5 | 32·646 |
| 13 | 7·015 | 29 | 15·649 | 45 | 24·283 | 61 | 32·916 |
| 13·5 | 7·285 | 29·5 | 15·919 | 45·5 | 24·552 | 61·5 | 33·186 |
| 14 | 7·555 | 30 | 16·188 | 46 | 24·822 | 62 | 33·456 |
| 14·5 | 7·824 | 30·5 | 16·458 | 46·5 | 25·092 | 62·5 | 33·726 |
| 15 | 8·094 | 31 | 16·728 | 47 | 25·362 | 63 | 33·996 |
| 15·5 | 8·364 | 31·5 | 16·998 | 47·5 | 25·632 | 63·5 | 34·266 |
| 16 | 8·634 | 32 | 17·268 | 48 | 25·901 | 64 | 34·536 |
| 16·5 | 8·904 | 32·5 | 17·538 | 48·5 | 26·171 | 64·5 | 34·806 |
| 17 | 9·173 | 33 | 17·807 | 49 | 26·441 | 65 | 35·076 |
| 17·5 | 9·443 | 33·5 | 18·077 | 49·5 | 26·711 | 65·5 | 35·345 |
| 18 | 9·713 | 34 | 18·347 | 50 | 26·981 | 66 | 35·614 |
| 18·5 | 9·983 | 34·5 | 18·617 | 50·5 | 27·251 | 66·5 | 35·884 |
| 19 | 10·253 | 35 | 18·887 | 51 | 27·520 | 67 | 36·154 |
| 19·5 | 10·523 | 35·5 | 19·156 | 51·5 | 27·790 | 67·5 | 36·424 |
| 20 | 10·792 | 36 | 19·426 | 52 | 28·060 | 68 | 36·694 |
| 20·5 | 11·062 | 36·5 | 19·696 | 52·5 | 28·330 | 68·5 | 36·964 |
| 21 | 11·332 | 37 | 19·966 | 53 | 28·600 | 69 | 37·234 |
| 21·5 | 11·602 | 37·5 | 20·235 | 53·5 | 28·869 | 69·5 | 37·504 |
| 22 | 11·872 | 38 | 20·505 | 54 | 29·139 | 70 | 37·774 |
| 22·5 | 12·141 | 38·5 | 20·775 | 54·5 | 29·409 | 70·5 | 38·044 |
| 23 | 12·411 | 39 | 21·045 | 55 | 29·679 | 71 | 38·314 |

$\frac{1}{10}$ Kilometre per hour = ·05396 of an Admiralty knot.
$\frac{1}{100}$ ,, ,, ,, = ·005396 ,, ,,
1 Admiralty knot = 6080 ft. per hour.
1 Statute mile = 5280 ft.
1 Kilometre = 3280·8992 ft.

## Table CLXXXII.—Millimetres and Inches.

| Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. |
|---|---|---|---|---|---|---|---|---|---|
| 1 | ·03937 | 41 | 1·6142 | 81 | 3·1890 | 121 | 4·7639 | 161 | 6·3387 |
| 2 | ·07874 | 42 | 1·6536 | 82 | 3·2284 | 122 | 4·8032 | 162 | 6·3781 |
| 3 | ·11811 | 43 | 1·6929 | 83 | 3·2678 | 123 | 4·8426 | 163 | 6·4174 |
| 4 | ·15748 | 44 | 1·7323 | 84 | 3·3071 | 124 | 4·8820 | 164 | 6·4568 |
| 5 | ·19685 | 45 | 1·7717 | 85 | 3·3465 | 125 | 4·9214 | 165 | 6·4962 |
| 6 | ·23622 | 46 | 1·8110 | 86 | 3·3859 | 126 | 4·9607 | 166 | 6·5356 |
| 7 | ·27560 | 47 | 1·8504 | 87 | 3·4252 | 127 | 5·0001 | 167 | 6·5749 |
| 8 | ·31497 | 48 | 1·8898 | 88 | 3·4646 | 128 | 5·0395 | 168 | 6·6143 |
| 9 | ·35434 | 49 | 1·9292 | 89 | 3·5040 | 129 | 5·0788 | 169 | 6·6537 |
| 10 | ·3937 | 50 | 1·9685 | 90 | 3·5434 | 130 | 5·1182 | 170 | 6·6930 |
| 11 | ·4331 | 51 | 2·0079 | 91 | 3·5827 | 131 | 5·1576 | 171 | 6·7324 |
| 12 | ·4724 | 52 | 2·0473 | 92 | 3·6221 | 132 | 5·1969 | 172 | 6·7718 |
| 13 | ·5118 | 53 | 2·0866 | 93 | 3·6614 | 133 | 5·2363 | 173 | 6·8111 |
| 14 | ·5512 | 54 | 2·1260 | 94 | 3·7008 | 134 | 5·2757 | 174 | 6·8505 |
| 15 | ·5906 | 55 | 2·1654 | 95 | 3·7402 | 135 | 5·3150 | 175 | 6·8899 |
| 16 | ·6299 | 56 | 2·2048 | 96 | 3·7796 | 136 | 5·3544 | 176 | 6·9293 |
| 17 | ·6693 | 57 | 2·2441 | 97 | 3·8190 | 137 | 5·3938 | 177 | 6·9686 |
| 18 | ·7087 | 58 | 2·2835 | 98 | 3·8583 | 138 | 5·4332 | 178 | 7·0080 |
| 19 | ·7480 | 59 | 2·3229 | 99 | 3·8977 | 139 | 5·4725 | 179 | 7·0474 |
| 20 | ·7874 | 60 | 2·3622 | 100 | 3·9371 | 140 | 5·5119 | 180 | 7·0867 |
| 21 | ·8268 | 61 | 2·4016 | 101 | 3·9764 | 141 | 5·5513 | 181 | 7·1261 |
| 22 | ·8662 | 62 | 2·4410 | 102 | 4·0158 | 142 | 5·5906 | 182 | 7·1655 |
| 23 | ·9055 | 63 | 2·4804 | 103 | 4·0552 | 143 | 5·6300 | 183 | 7·2048 |
| 24 | ·9449 | 64 | 2·5197 | 104 | 4·0946 | 144 | 5·6694 | 184 | 7·2442 |
| 25 | ·9843 | 65 | 2·5591 | 105 | 4·1339 | 145 | 5·7088 | 185 | 7·2836 |
| 26 | 1·0236 | 66 | 2·5985 | 106 | 4·1733 | 146 | 5·7481 | 186 | 7·3230 |
| 27 | 1·0630 | 67 | 2·6378 | 107 | 4·2127 | 147 | 5·7875 | 187 | 7·3623 |
| 28 | 1·1024 | 68 | 2·6772 | 108 | 4·2520 | 148 | 5·8269 | 188 | 7·4017 |
| 29 | 1·1418 | 69 | 2·7166 | 109 | 4·2914 | 149 | 5·8662 | 189 | 7·4411 |
| 30 | 1·1811 | 70 | 2·7560 | 110 | 4·3308 | 150 | 5·9056 | 190 | 7·4804 |
| 31 | 1·2205 | 71 | 2·7953 | 111 | 4·3702 | 151 | 5·9450 | 191 | 7·5198 |
| 32 | 1·2599 | 72 | 2·8347 | 112 | 4·4095 | 152 | 5·9844 | 192 | 7·5592 |
| 33 | 1·2992 | 73 | 2·8741 | 113 | 4·4489 | 153 | 6·0237 | 193 | 7·5986 |
| 34 | 1·3386 | 74 | 2·9134 | 114 | 4·4883 | 154 | 6·0631 | 194 | 7·6379 |
| 35 | 1·3780 | 75 | 2·9528 | 115 | 4·5276 | 155 | 6·1025 | 195 | 7·6773 |
| 36 | 1·4173 | 76 | 2·9922 | 116 | 4·5670 | 156 | 6·1418 | 196 | 7·7167 |
| 37 | 1·4567 | 77 | 3·0316 | 117 | 4·6064 | 157 | 6·1812 | 197 | 7·7560 |
| 38 | 1·4961 | 78 | 3·0709 | 118 | 4·6458 | 158 | 6·2206 | 198 | 7·7954 |
| 39 | 1·5355 | 79 | 3·1103 | 119 | 4·6851 | 159 | 6·2600 | 199 | 7·8348 |
| 40 | 1·5748 | 80 | 3·1497 | 120 | 4·7245 | 160 | 6·2993 | 200 | 7·8742 |

## Table CLXXXII.—Millimetres and Inches—*continued.*

| Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. |
|---|---|---|---|---|---|---|---|---|---|
| 201 | 7·9135 | 241 | 9·4884 | 281 | 11·0632 | 321 | 12·6380 | 361 | 14·2128 |
| 202 | 7·9529 | 242 | 9·5277 | 282 | 11·1026 | 322 | 12·6774 | 362 | 14·2522 |
| 203 | 7·9923 | 243 | 9·5671 | 283 | 11·1419 | 323 | 12·7168 | 363 | 14·2916 |
| 204 | 8·0316 | 244 | 9·6065 | 284 | 11·1813 | 324 | 12·7561 | 364 | 14·3310 |
| 205 | 8·0710 | 245 | 9·6458 | 285 | 11·2207 | 325 | 12·7955 | 365 | 14·3703 |
| 206 | 8·1104 | 246 | 9·6852 | 286 | 11·2600 | 326 | 12·8349 | 366 | 14·4097 |
| 207 | 8·1498 | 247 | 9·7246 | 287 | 11·2994 | 327 | 12·8742 | 367 | 14·4491 |
| 208 | 8·1891 | 248 | 9·7640 | 288 | 11·3388 | 328 | 12·9136 | 368 | 14·4884 |
| 209 | 8·2285 | 249 | 9·8033 | 289 | 11·3782 | 329 | 12·9530 | 369 | 14·5278 |
| 210 | 8·2679 | 250 | 9·8427 | 290 | 11·4175 | 330 | 12·9924 | 370 | 14·5672 |
| 211 | 8·3072 | 251 | 9·8821 | 291 | 11·4569 | 331 | 13·0317 | 371 | 14·6066 |
| 212 | 8·3466 | 252 | 9·9214 | 292 | 11·4963 | 332 | 13·0711 | 372 | 14·6459 |
| 213 | 8·3860 | 253 | 9·9608 | 293 | 11·5356 | 333 | 13·1105 | 373 | 14·6853 |
| 214 | 8·4254 | 254 | 10·0002 | 294 | 11·5750 | 334 | 13·1498 | 374 | 14·7247 |
| 215 | 8·4647 | 255 | 10·0396 | 295 | 11·6144 | 335 | 13·1892 | 375 | 14·7640 |
| 216 | 8·5041 | 256 | 10·0789 | 296 | 11·6538 | 336 | 13·2286 | 376 | 14·8034 |
| 217 | 8·5435 | 257 | 10·1183 | 297 | 11·6931 | 337 | 13·2680 | 377 | 14·8428 |
| 218 | 8·5828 | 258 | 10·1577 | 298 | 11·7325 | 338 | 13·3073 | 378 | 14·8822 |
| 219 | 8·6222 | 259 | 10·1970 | 299 | 11·7719 | 339 | 13·3467 | 379 | 14·9215 |
| 220 | 8·6616 | 260 | 10·2364 | 300 | 11·8112 | 340 | 13·3861 | 380 | 14·9609 |
| 221 | 8·7009 | 261 | 10·2758 | 301 | 11·8506 | 341 | 13·4254 | 381 | 15·0003 |
| 222 | 8·7403 | 262 | 10·3151 | 302 | 11·8900 | 342 | 13·4648 | 382 | 15·0396 |
| 223 | 8·7797 | 263 | 10·3545 | 303 | 11·9294 | 343 | 13·5042 | 383 | 15·0790 |
| 224 | 8·8190 | 264 | 10·3939 | 304 | 11·9687 | 344 | 13·5436 | 384 | 15·1184 |
| 225 | 8·8584 | 265 | 10·4333 | 305 | 12·0081 | 345 | 13·5829 | 385 | 15·1578 |
| 226 | 8·8978 | 266 | 10·4726 | 306 | 12·0475 | 346 | 13·6223 | 386 | 15·1971 |
| 227 | 8·9372 | 267 | 10·5120 | 307 | 12·0868 | 347 | 13·6617 | 387 | 15·2365 |
| 228 | 8·9765 | 268 | 10·5514 | 308 | 12·1262 | 348 | 13·7010 | 388 | 15·2759 |
| 229 | 9·0159 | 269 | 10·5907 | 309 | 12·1656 | 349 | 13·7404 | 389 | 15·3152 |
| 230 | 9·0553 | 270 | 10·6301 | 310 | 12·2049 | 350 | 13·7798 | 390 | 15·3546 |
| 231 | 9·0946 | 271 | 10·6695 | 311 | 12·2443 | 351 | 13·8192 | 391 | 15·3940 |
| 232 | 9·1340 | 272 | 10·7088 | 312 | 12·2837 | 352 | 13·8585 | 392 | 15·4334 |
| 233 | 9·1734 | 273 | 10·7482 | 313 | 12·3231 | 353 | 13·8979 | 393 | 15·4727 |
| 234 | 9·2128 | 274 | 10·7876 | 314 | 12·3624 | 354 | 13·9373 | 394 | 15·5121 |
| 235 | 9·2521 | 275 | 10·8270 | 315 | 12·4018 | 355 | 13·9766 | 395 | 15·5515 |
| 236 | 9·2915 | 276 | 10·8663 | 316 | 12·4412 | 356 | 14·0160 | 396 | 15·5908 |
| 237 | 9·3309 | 277 | 10·9057 | 317 | 12·4805 | 357 | 14·0554 | 397 | 15·6302 |
| 238 | 9·3702 | 278 | 10·9451 | 318 | 12·5199 | 358 | 14·0947 | 398 | 15·6696 |
| 239 | 9·4096 | 279 | 10·9844 | 319 | 12·5593 | 359 | 14·1341 | 399 | 15·7089 |
| 240 | 9·4490 | 280 | 11·0238 | 320 | 12·5986 | 360 | 14·1735 | 400 | 15·7483 |

## Table CLXXXII.—Millimetres and Inches—*continued.*

| Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. |
|---|---|---|---|---|---|---|---|---|---|
| 401 | 15·7877 | 441 | 17·3625 | 481 | 18·9374 | 521 | 20·5122 | 561 | 22·0870 |
| 402 | 15·8271 | 442 | 17·4019 | 482 | 18·9767 | 522 | 20·5516 | 562 | 22·1264 |
| 403 | 15·8664 | 443 | 17·4413 | 483 | 19·0161 | 523 | 20·5909 | 563 | 22·1658 |
| 404 | 15·9058 | 444 | 17·4806 | 484 | 19·0555 | 524 | 20·6303 | 564 | 22·2051 |
| 405 | 15·9452 | 445 | 17·5200 | 485 | 19·0948 | 525 | 20·6697 | 565 | 22·2445 |
| 406 | 15·9845 | 446 | 17·5594 | 486 | 19·1342 | 526 | 20·7090 | 566 | 22·2839 |
| 407 | 16·0239 | 447 | 17·5987 | 487 | 19·1736 | 527 | 20·7484 | 567 | 22·3232 |
| 408 | 16·0633 | 448 | 17·6381 | 488 | 19·2130 | 528 | 20·7878 | 568 | 22·3626 |
| 409 | 16·1026 | 449 | 17·6775 | 489 | 19·2523 | 529 | 20·8272 | 569 | 22·4020 |
| 410 | 16·1420 | 450 | 17·7169 | 490 | 19·2917 | 530 | 20·8665 | 570 | 22·4414 |
| 411 | 16·1814 | 451 | 17·7562 | 491 | 19·3311 | 531 | 20·9059 | 571 | 22·4807 |
| 412 | 16·2208 | 452 | 17·7956 | 492 | 19·3704 | 532 | 20·9453 | 572 | 22·5201 |
| 413 | 16·2601 | 453 | 17·8350 | 493 | 19·4098 | 533 | 20·9846 | 573 | 22·5595 |
| 414 | 16·2995 | 454 | 17·8743 | 494 | 19·4492 | 534 | 21·0240 | 574 | 22·5988 |
| 415 | 16·3389 | 455 | 17·9137 | 495 | 19·4885 | 535 | 21·0634 | 575 | 22·6382 |
| 416 | 16·3782 | 456 | 17·9531 | 496 | 19·5279 | 536 | 21·1027 | 576 | 22·6776 |
| 417 | 16·4176 | 457 | 17·9924 | 497 | 19·5673 | 537 | 21·1421 | 577 | 22·7170 |
| 418 | 16·4570 | 458 | 18·0318 | 498 | 19·6066 | 538 | 21·1815 | 578 | 22·7563 |
| 419 | 16·4964 | 459 | 18·0712 | 499 | 19·6460 | 539 | 21·2209 | 579 | 22·7957 |
| 420 | 16·5357 | 460 | 18·1106 | 500 | 19·6854 | 540 | 21·2602 | 580 | 22·8351 |
| 421 | 16·5751 | 461 | 18·1499 | 501 | 19·7248 | 541 | 21·2996 | 581 | 22·8744 |
| 422 | 16·6145 | 462 | 18·1893 | 502 | 19·7641 | 542 | 21·3390 | 582 | 22·9138 |
| 423 | 16·6538 | 463 | 18·2287 | 503 | 19·8035 | 543 | 21·3783 | 583 | 22·9532 |
| 424 | 16·6932 | 464 | 18·2680 | 504 | 19·8429 | 544 | 21·4177 | 584 | 22·9925 |
| 425 | 16·7326 | 465 | 18·3074 | 505 | 19·8822 | 545 | 21·4571 | 585 | 23·0319 |
| 426 | 16·7720 | 466 | 18·3468 | 506 | 19·9216 | 546 | 21·4964 | 586 | 23·0713 |
| 427 | 16·8113 | 467 | 18·3862 | 507 | 19·9610 | 547 | 21·5358 | 587 | 23·1106 |
| 428 | 16·8507 | 468 | 18·4255 | 508 | 20·0004 | 548 | 21·5752 | 588 | 23·1500 |
| 429 | 16·8901 | 469 | 18·4649 | 509 | 20·0397 | 549 | 21·6146 | 589 | 23·1894 |
| 430 | 16·9294 | 470 | 18·5043 | 510 | 20·0791 | 550 | 21·6539 | 590 | 23·2288 |
| 431 | 16·9688 | 471 | 18·5436 | 511 | 20·1185 | 551 | 21·6933 | 591 | 23·2681 |
| 432 | 17·0082 | 472 | 18·5830 | 512 | 20·1578 | 552 | 21·7327 | 592 | 23·3075 |
| 433 | 17·0476 | 473 | 18·6224 | 513 | 20·1972 | 553 | 21·7720 | 593 | 23·3469 |
| 434 | 17·0869 | 474 | 18·6617 | 514 | 20·2366 | 554 | 21·8114 | 594 | 23·3862 |
| 435 | 17·1263 | 475 | 18·7011 | 515 | 20·2760 | 555 | 21·8508 | 595 | 23·4256 |
| 436 | 17·1657 | 476 | 18·7405 | 516 | 20·3153 | 556 | 21·8902 | 596 | 23·4650 |
| 437 | 17·2050 | 477 | 18·7799 | 517 | 20·3547 | 557 | 21·9295 | 597 | 23·5044 |
| 438 | 17·2444 | 478 | 18·8192 | 518 | 20·3941 | 558 | 21·9689 | 598 | 23·5437 |
| 439 | 17·2838 | 479 | 18·8586 | 519 | 20·4334 | 559 | 22·0083 | 599 | 23·5831 |
| 440 | 17·3232 | 480 | 18·8980 | 520 | 20·4728 | 560 | 22·0476 | 600 | 23·6225 |

## Table CLXXXII.—Millimetres and Inches—*continued.*

| Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. |
|---|---|---|---|---|---|---|---|---|---|
| 601 | 23·6618 | 641 | 25·2367 | 681 | 26·8115 | 721 | 28·3863 | 761 | 29·9612 |
| 602 | 23·7012 | 642 | 25·2760 | 682 | 26·8509 | 722 | 28·4257 | 762 | 30·0005 |
| 603 | 23·7406 | 643 | 25·3154 | 683 | 26·8902 | 723 | 28·4651 | 763 | 30·0399 |
| 604 | 23·7800 | 644 | 25·3548 | 684 | 26·9296 | 724 | 28·5044 | 764 | 30·0793 |
| 605 | 23·8193 | 645 | 25·3942 | 685 | 26·9690 | 725 | 28·5438 | 765 | 30·1187 |
| 606 | 23·8587 | 646 | 25·4335 | 686 | 27·0084 | 726 | 28·5832 | 766 | 30·1580 |
| 607 | 23·8981 | 647 | 25·4729 | 687 | 27·0477 | 727 | 28·6226 | 767 | 30·1974 |
| 608 | 23·9374 | 648 | 25·5123 | 688 | 27·0871 | 728 | 28·6619 | 768 | 30·2368 |
| 609 | 23·9768 | 649 | 25·5516 | 689 | 27·1265 | 729 | 28·7013 | 769 | 30·2761 |
| 610 | 24·0162 | 650 | 25·5910 | 690 | 27·1658 | 730 | 28·7407 | 770 | 30·3155 |
| 611 | 24·0556 | 651 | 25·6304 | 691 | 27·2052 | 731 | 28·7800 | 771 | 30·3549 |
| 612 | 24·0949 | 652 | 25·6698 | 692 | 27·2446 | 732 | 28·8194 | 772 | 30·3942 |
| 613 | 24·1343 | 653 | 25·7091 | 693 | 27·2840 | 733 | 28·8588 | 773 | 30·4336 |
| 614 | 24·1737 | 654 | 25·7485 | 694 | 27·3233 | 734 | 28·8982 | 774 | 30·4730 |
| 615 | 24·2130 | 655 | 25·7879 | 695 | 27·3627 | 735 | 28·9375 | 775 | 30·5124 |
| 616 | 24·2524 | 656 | 25·8272 | 696 | 27·4021 | 736 | 28·9769 | 776 | 30·5517 |
| 617 | 24·2918 | 657 | 25·8666 | 697 | 27·4414 | 737 | 29·0163 | 777 | 30·5911 |
| 618 | 24·3312 | 658 | 25·9060 | 698 | 27·4808 | 738 | 29·0556 | 778 | 30·6305 |
| 619 | 24·3705 | 659 | 25·9454 | 699 | 27·5202 | 739 | 29·0950 | 779 | 30·6698 |
| 620 | 24·4099 | 660 | 25·9847 | 700 | 27·5596 | 740 | 29·1344 | 780 | 30·7092 |
| 621 | 24·4493 | 661 | 26·0241 | 701 | 27·5989 | 741 | 29·1738 | 781 | 30·7486 |
| 622 | 24·4886 | 662 | 26·0635 | 702 | 27·6383 | 742 | 29·2131 | 782 | 30·7880 |
| 623 | 24·5280 | 663 | 26·1028 | 703 | 27·6777 | 743 | 29·2525 | 783 | 30·8273 |
| 624 | 24·5674 | 664 | 26·1422 | 704 | 27·7170 | 744 | 29·2919 | 784 | 30·8667 |
| 625 | 24·6068 | 665 | 26·1816 | 705 | 27·7564 | 745 | 29·3312 | 785 | 30·9061 |
| 626 | 24·6461 | 666 | 26·2210 | 706 | 27·7958 | 746 | 29·3706 | 786 | 30·9454 |
| 627 | 24·6855 | 667 | 26·2603 | 707 | 27·8352 | 747 | 29·4100 | 787 | 30·9848 |
| 628 | 24·7249 | 668 | 26·2997 | 708 | 27·8745 | 748 | 29·4494 | 788 | 31·0242 |
| 629 | 24·7642 | 669 | 26·3391 | 709 | 27·9139 | 749 | 29·4887 | 789 | 31·0636 |
| 630 | 24·8036 | 670 | 26·3784 | 710 | 27·9533 | 750 | 29·5281 | 790 | 31·1029 |
| 631 | 24·8430 | 671 | 26·4178 | 711 | 27·9926 | 751 | 29·5675 | 791 | 31·1423 |
| 632 | 24·8823 | 672 | 26·4572 | 712 | 28·0320 | 752 | 29·6068 | 792 | 31·1817 |
| 633 | 24·9217 | 673 | 26·4965 | 713 | 28·0714 | 753 | 29·6462 | 793 | 31·2210 |
| 634 | 24·9611 | 674 | 26·5359 | 714 | 28·1108 | 754 | 29·6856 | 794 | 31·2604 |
| 635 | 25·0004 | 675 | 26·5753 | 715 | 28·1501 | 755 | 29·7250 | 795 | 31·2998 |
| 636 | 25·0398 | 676 | 26·6147 | 716 | 28·1895 | 756 | 29·7643 | 796 | 31·3392 |
| 637 | 25·0792 | 677 | 26·6540 | 717 | 28·2289 | 757 | 29·8037 | 797 | 31·3785 |
| 638 | 25·1186 | 678 | 26·6934 | 718 | 28·2682 | 758 | 29·8431 | 798 | 31·4179 |
| 639 | 25·1579 | 679 | 26·7328 | 719 | 28·3076 | 759 | 29·8824 | 799 | 31·4573 |
| 640 | 25·1973 | 680 | 26·7721 | 720 | 28·3470 | 760 | 29·9218 | 800 | 31·4966 |

## Table CLXXXII.—Millimetres and Inches—*continued.*

| Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. | Milli-metres. | Inches. |
|---|---|---|---|---|---|---|---|---|---|
| 801 | 31·5360 | 841 | 33·1108 | 881 | 34·6857 | 921 | 36·2605 | 961 | 37·8253 |
| 802 | 31·5754 | 842 | 33·1502 | 882 | 34·7250 | 922 | 36·2999 | 962 | 37·8747 |
| 803 | 31·6148 | 843 | 33·1896 | 883 | 34·7644 | 923 | 36·3392 | 963 | 37·9141 |
| 804 | 31·6541 | 844 | 33·2290 | 884 | 34·8038 | 924 | 36·3786 | 964 | 37·9534 |
| 805 | 31·6935 | 845 | 33·2683 | 885 | 34·8432 | 925 | 36·4180 | 965 | 37·9928 |
| 806 | 31·7329 | 846 | 33·3077 | 886 | 34·8825 | 926 | 36·4574 | 966 | 38·0322 |
| 807 | 31·7722 | 847 | 33·3471 | 887 | 34·9219 | 927 | 36·4967 | 967 | 38·0716 |
| 808 | 31·8116 | 848 | 33·3864 | 888 | 34·9613 | 928 | 36·5361 | 968 | 38·1109 |
| 809 | 31·8510 | 849 | 33·4258 | 889 | 35·0006 | 929 | 36·5755 | 969 | 38·1503 |
| 810 | 31·8903 | 850 | 33·4652 | 890 | 35·0400 | 930 | 36·6148 | 970 | 38·1897 |
| 811 | 31·9297 | 851 | 33·5046 | 891 | 35·0794 | 931 | 36·6542 | 971 | 38·2290 |
| 812 | 31·9691 | 852 | 33·5439 | 892 | 35·1188 | 932 | 36·6936 | 972 | 38·2684 |
| 813 | 32·0085 | 853 | 33·5833 | 893 | 35·1581 | 933 | 36·7330 | 973 | 38·3078 |
| 814 | 32·0478 | 854 | 33·6227 | 894 | 35·1975 | 934 | 36·7723 | 974 | 38·3472 |
| 815 | 32·0872 | 855 | 33·6620 | 895 | 35·2369 | 935 | 36·8117 | 975 | 38·3865 |
| 816 | 32·1266 | 856 | 33·7014 | 896 | 35·2762 | 936 | 36·8511 | 976 | 38·4259 |
| 817 | 32·1659 | 857 | 33·7408 | 897 | 35·3156 | 937 | 36·8904 | 977 | 38·4653 |
| 818 | 32·2053 | 858 | 33·7801 | 898 | 35·3550 | 938 | 36·9298 | 978 | 38·5046 |
| 819 | 32·2447 | 859 | 33·8195 | 899 | 35·3943 | 939 | 36·9692 | 979 | 38·5440 |
| 820 | 32·2840 | 860 | 33·8589 | 900 | 35·4337 | 940 | 37·0086 | 980 | 38·5834 |
| 821 | 32·3234 | 861 | 33·8982 | 901 | 35·4731 | 941 | 37·0479 | 981 | 38·6228 |
| 822 | 32·3628 | 862 | 33·9376 | 902 | 35·5125 | 942 | 37·0873 | 982 | 38·6621 |
| 823 | 32·4022 | 863 | 33·9770 | 903 | 35·5518 | 943 | 37·1267 | 983 | 38·7015 |
| 824 | 32·4415 | 864 | 34·0164 | 904 | 35·5912 | 944 | 37·1660 | 984 | 38·7409 |
| 825 | 32·4809 | 865 | 34·0557 | 905 | 35·6306 | 945 | 37·2054 | 985 | 38·7802 |
| 826 | 32·5203 | 866 | 34·0951 | 906 | 35·6699 | 946 | 37·2448 | 986 | 38·8196 |
| 827 | 32·5596 | 867 | 34·1345 | 907 | 35·7093 | 947 | 37·2841 | 987 | 38·8590 |
| 828 | 32·5990 | 868 | 34·1738 | 908 | 35·7487 | 948 | 37·3235 | 988 | 38·8984 |
| 829 | 32·6384 | 869 | 34·2132 | 909 | 35·7880 | 949 | 37·3629 | 989 | 38·9377 |
| 830 | 32·6778 | 870 | 34·2526 | 910 | 35·8274 | 950 | 37·4023 | 990 | 38·9771 |
| 831 | 32·7171 | 871 | 34·2920 | 911 | 35·8668 | 951 | 37·4416 | 991 | 39·0165 |
| 832 | 32·7565 | 872 | 34·3313 | 912 | 35·9062 | 952 | 37·4810 | 992 | 39·0558 |
| 833 | 32·7959 | 873 | 34·3707 | 913 | 35·9455 | 953 | 37·5204 | 993 | 39·0952 |
| 834 | 32·8352 | 874 | 34·4101 | 914 | 35·9849 | 954 | 37·5597 | 994 | 39·1346 |
| 835 | 32·8746 | 875 | 34·4494 | 915 | 36·0243 | 955 | 37·5991 | 995 | 39·1739 |
| 836 | 32·9140 | 876 | 34·4888 | 916 | 36·0636 | 956 | 37·6385 | 996 | 39·2133 |
| 837 | 32·9534 | 877 | 34·5282 | 917 | 36·1030 | 957 | 37·6778 | 997 | 39·2527 |
| 838 | 32·9927 | 878 | 34·5676 | 918 | 36·1424 | 958 | 37·7172 | 998 | 39·2920 |
| 839 | 33·0321 | 879 | 34·6069 | 919 | 36·1818 | 959 | 37·7566 | 999 | 39·3314 |
| 840 | 33·0715 | 880 | 34·6463 | 920 | 36·2211 | 960 | 37·7960 | 1000 | 39·3708 |

965 mm = 37·9928 ins.: 9·65 metres = 379·928 ins.: 96·5 metres = 3799·28 ins.: &c.

## Table CLXXXIIa.—English Feet and French Metres.

| Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1·0 | 0·305 | 13·0 | 3·965 | 25 | 7·625 | 49 | 14·945 | 73 | 22·265 | 97 | 29·585 | 121 | 36·905 |
| 1·5 | 0·458 | 13·5 | 4·118 | 26 | 7·930 | 50 | 15·250 | 74 | 22·570 | 98 | 29·890 | 122 | 37·210 |
| 2·0 | 0·610 | 14·0 | 4·270 | 27 | 8·235 | 51 | 15·555 | 75 | 22·875 | 99 | 30·195 | 123 | 37·515 |
| 2·5 | 0·762 | 14·5 | 4·423 | 28 | 8·540 | 52 | 15·860 | 76 | 23·180 | 100 | 30·500 | 124 | 37·820 |
| 3·0 | 0·915 | 15·0 | 4·575 | 29 | 8·845 | 53 | 16·165 | 77 | 23·485 | 101 | 30·805 | 125 | 38·125 |
| 3·5 | 1·068 | 15·5 | 4·728 | 30 | 9·150 | 54 | 16·470 | 78 | 23·790 | 102 | 31·110 | 126 | 38·430 |
| 4·0 | 1·220 | 16·0 | 4·880 | 31 | 9·455 | 55 | 16·775 | 79 | 24·095 | 103 | 31·415 | 127 | 38·735 |
| 4·5 | 1·373 | 16·5 | 5·033 | 32 | 9·760 | 56 | 17·080 | 80 | 24·400 | 104 | 31·720 | 128 | 39·040 |
| 5·0 | 1·525 | 17·0 | 5·185 | 33 | 10·065 | 57 | 17·385 | 81 | 24·705 | 105 | 32·025 | 129 | 39·345 |
| 5·5 | 1·678 | 17·5 | 5·338 | 34 | 10·370 | 58 | 17·690 | 82 | 25·010 | 106 | 32·330 | 130 | 39·650 |
| 6·0 | 1·830 | 18·0 | 5·490 | 35 | 10·675 | 59 | 17·995 | 83 | 25·315 | 107 | 32·635 | 131 | 39·955 |
| 6·5 | 1·983 | 18·5 | 5·643 | 36 | 10·980 | 60 | 18·300 | 84 | 25·620 | 108 | 32·940 | 132 | 40·260 |
| 7·0 | 2·135 | 19·0 | 5·795 | 37 | 11·285 | 61 | 18·605 | 85 | 25·925 | 109 | 33·245 | 133 | 40·565 |
| 7·5 | 2·288 | 19·5 | 5·948 | 38 | 11·590 | 62 | 18·910 | 86 | 26·230 | 110 | 33·550 | 134 | 40·870 |
| 8·0 | 2·440 | 20·0 | 6·100 | 39 | 11·895 | 63 | 19·215 | 87 | 26·535 | 111 | 33·855 | 135 | 41·175 |
| 8·5 | 2·593 | 20·5 | 6·253 | 40 | 12·200 | 64 | 19·520 | 88 | 26·840 | 112 | 34·160 | 136 | 41·480 |
| 9·0 | 2·745 | 21·0 | 6·405 | 41 | 12·505 | 65 | 19·825 | 89 | 27·145 | 113 | 34·465 | 137 | 41·785 |
| 9·5 | 2·898 | 21·5 | 6·558 | 42 | 12·810 | 66 | 20·130 | 90 | 27·450 | 114 | 34·770 | 138 | 42·090 |
| 10·0 | 3·050 | 22·0 | 6·710 | 43 | 13·115 | 67 | 20·435 | 91 | 27·755 | 115 | 35·075 | 139 | 42·395 |
| 10·5 | 3·203 | 22·5 | 6·863 | 44 | 13·420 | 68 | 20·740 | 92 | 28·060 | 116 | 35·380 | 140 | 42·700 |
| 11·0 | 3·355 | 23·0 | 7·015 | 45 | 13·725 | 69 | 21·045 | 93 | 28·365 | 117 | 35·685 | 141 | 43·005 |
| 11·5 | 3·508 | 23·5 | 7·168 | 46 | 14·030 | 70 | 21·350 | 94 | 28·670 | 118 | 35·990 | 142 | 43·310 |
| 12·0 | 3·660 | 24·0 | 7·320 | 47 | 14·335 | 71 | 21·655 | 95 | 28·975 | 119 | 36·295 | 143 | 43·615 |
| 12·5 | 3·813 | 24·5 | 7·473 | 48 | 14·640 | 72 | 21·960 | 96 | 29·280 | 120 | 36·600 | 144 | 43·920 |

## Table CLXXXIIa.—English Feet and French Metres—*continued.*

| Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 145 | 44·225 | 169 | 51·545 | 193 | 58·865 | 217 | 66·185 | 241 | 73·505 | 265 | 80·825 | 289 | 88·145 |
| 146 | 44·530 | 170 | 51·850 | 194 | 59·170 | 218 | 66·490 | 242 | 73·810 | 266 | 81·130 | 290 | 88·450 |
| 147 | 44·835 | 171 | 52·155 | 195 | 59·475 | 219 | 66·795 | 243 | 74·115 | 267 | 81·435 | 291 | 88·755 |
| 148 | 45·140 | 172 | 52·460 | 196 | 59·780 | 220 | 67·100 | 244 | 74·420 | 268 | 81·740 | 292 | 89·060 |
| 149 | 45·445 | 173 | 52·765 | 197 | 60·085 | 221 | 67·405 | 245 | 74·725 | 269 | 82·045 | 293 | 89·365 |
| 150 | 45·750 | 174 | 53·070 | 198 | 60·390 | 222 | 67·710 | 246 | 75·030 | 270 | 82·350 | 294 | 89·670 |
| 151 | 46·055 | 175 | 53·375 | 199 | 60·695 | 223 | 68·015 | 247 | 75·335 | 271 | 82·655 | 295 | 89·975 |
| 152 | 46·360 | 176 | 53·680 | 200 | 61·000 | 224 | 68·320 | 248 | 75·640 | 272 | 82·960 | 296 | 90·280 |
| 153 | 46·665 | 177 | 53·985 | 201 | 61·305 | 225 | 68·625 | 249 | 75·945 | 273 | 83·265 | 297 | 90·585 |
| 154 | 46·970 | 178 | 54·290 | 202 | 61·610 | 226 | 68·930 | 250 | 76·250 | 274 | 83·570 | 298 | 90·890 |
| 155 | 47·275 | 179 | 54·595 | 203 | 61·915 | 227 | 69·235 | 251 | 76·555 | 275 | 83·875 | 299 | 91·195 |
| 156 | 47·580 | 180 | 54·900 | 204 | 62·220 | 228 | 69·540 | 252 | 76·860 | 276 | 84·180 | 300 | 91·500 |
| 157 | 47·885 | 181 | 55·205 | 205 | 62·525 | 229 | 69·845 | 253 | 77·165 | 277 | 84·485 | 301 | 91·805 |
| 158 | 48·190 | 182 | 55·510 | 206 | 62·830 | 230 | 70·150 | 254 | 77·470 | 278 | 84·790 | 302 | 92·110 |
| 159 | 48·495 | 183 | 55·815 | 207 | 63·135 | 231 | 70·455 | 255 | 77·775 | 279 | 85·095 | 303 | 92·415 |
| 160 | 48·800 | 184 | 56·120 | 208 | 63·440 | 232 | 70·760 | 256 | 78·080 | 280 | 85·400 | 304 | 92·720 |
| 161 | 49·105 | 185 | 56·425 | 209 | 63·745 | 233 | 71·065 | 257 | 78·385 | 281 | 85·705 | 305 | 93·025 |
| 162 | 49·410 | 186 | 56·730 | 210 | 64·050 | 234 | 71·370 | 258 | 78·790 | 282 | 86·010 | 306 | 93·330 |
| 163 | 49·715 | 187 | 57·035 | 211 | 64·355 | 235 | 71·675 | 259 | 79·095 | 283 | 86·315 | 307 | 93·635 |
| 164 | 50·020 | 188 | 57·340 | 212 | 64·660 | 236 | 71·980 | 260 | 79·400 | 284 | 86·620 | 308 | 93·940 |
| 165 | 50·325 | 189 | 57·645 | 213 | 64·965 | 237 | 72·285 | 261 | 79·705 | 285 | 86·925 | 309 | 94·245 |
| 166 | 50·630 | 190 | 57·950 | 214 | 65·270 | 238 | 72·590 | 262 | 80·010 | 286 | 87·230 | 310 | 94·550 |
| 167 | 50·935 | 191 | 58·255 | 215 | 65·575 | 239 | 72·895 | 263 | 80·215 | 287 | 87·535 | 311 | 94·855 |
| 168 | 51·240 | 192 | 58·560 | 216 | 65·880 | 240 | 73·200 | 264 | 80·520 | 288 | 87·840 | 312 | 95·160 |

## Table CLXXXIIa.—English Feet and French Metres—*continued.*

| Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. | Feet. | Metres. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 313 | 95·465 | 337 | 102·79 | 361 | 110·11 | 385 | 117·43 | 409 | 124·75 | 433 | 132·07 | 457 | 139·39 |
| 314 | 95·770 | 338 | 103·09 | 362 | 110·41 | 386 | 117·73 | 410 | 125·05 | 434 | 132·37 | 458 | 139·69 |
| 315 | 96·075 | 339 | 103·40 | 363 | 110·72 | 387 | 118·04 | 411 | 125·36 | 435 | 132·68 | 459 | 140·00 |
| 316 | 96·380 | 340 | 103·70 | 364 | 111·02 | 388 | 118·34 | 412 | 125·66 | 436 | 132·98 | 460 | 140·30 |
| 317 | 96·685 | 341 | 104·07 | 365 | 111·33 | 389 | 118·65 | 413 | 125·97 | 437 | 133·29 | 461 | 140·61 |
| 318 | 96·990 | 342 | 104·31 | 366 | 111·63 | 390 | 118·95 | 414 | 126·27 | 438 | 133·59 | 462 | 140·91 |
| 319 | 97·295 | 343 | 104·62 | 367 | 111·94 | 391 | 119·26 | 415 | 126·58 | 439 | 133·90 | 463 | 141·22 |
| 320 | 97·600 | 344 | 104·92 | 368 | 112·24 | 392 | 119·56 | 416 | 126·88 | 440 | 134·20 | 464 | 141·52 |
| 321 | 97·905 | 345 | 105·23 | 369 | 112·55 | 393 | 119·87 | 417 | 127·19 | 441 | 134·51 | 465 | 141·83 |
| 322 | 98·210 | 346 | 105·53 | 370 | 112·85 | 394 | 120·17 | 418 | 127·49 | 442 | 134·81 | 466 | 142·13 |
| 323 | 98·515 | 347 | 105·83 | 371 | 113·16 | 395 | 120·48 | 419 | 127·80 | 443 | 135·12 | 467 | 142·44 |
| 324 | 98·820 | 348 | 106·13 | 372 | 113·46 | 396 | 120·78 | 420 | 128·10 | 444 | 135·42 | 468 | 142·74 |
| 325 | 99·125 | 349 | 106·43 | 373 | 113·77 | 397 | 121·09 | 421 | 128·41 | 445 | 135·73 | 469 | 143·05 |
| 326 | 99·430 | 350 | 106·75 | 374 | 114·07 | 398 | 121·39 | 422 | 121·71 | 446 | 136·03 | 470 | 143·35 |
| 327 | 99·735 | 351 | 107·06 | 375 | 114·38 | 399 | 121·70 | 423 | 129·02 | 447 | 136·34 | 471 | 143·66 |
| 328 | 100·04 | 352 | 107·36 | 376 | 114·68 | 400 | 122·00 | 424 | 129·32 | 448 | 136·64 | 472 | 143·96 |
| 329 | 100·35 | 353 | 107·67 | 377 | 114·99 | 401 | 122·31 | 425 | 129·63 | 449 | 136·95 | 473 | 144·27 |
| 330 | 100·65 | 354 | 107·97 | 378 | 115·29 | 402 | 122·61 | 426 | 129·93 | 450 | 137·25 | 474 | 144·57 |
| 331 | 100·96 | 355 | 108·28 | 379 | 115·60 | 403 | 122·92 | 427 | 130·24 | 451 | 137·56 | 475 | 144·88 |
| 332 | 101·26 | 356 | 108·58 | 380 | 115·90 | 404 | 123·22 | 428 | 130·54 | 452 | 137·86 | 476 | 145·18 |
| 333 | 101·57 | 357 | 108·89 | 381 | 116·21 | 405 | 123·53 | 429 | 130·85 | 453 | 138·17 | 477 | 145·49 |
| 334 | 101·87 | 358 | 109·19 | 382 | 116·51 | 406 | 123·83 | 430 | 131·15 | 454 | 138·47 | 478 | 145·79 |
| 335 | 102·18 | 359 | 109·50 | 383 | 116·82 | 407 | 124·14 | 431 | 131·46 | 455 | 138·78 | 479 | 146·10 |
| 336 | 102·48 | 360 | 109·80 | 384 | 117·12 | 408 | 124·44 | 432 | 131·76 | 456 | 139·08 | 480 | 146·40 |

## Table CLXXXIII.—Decimal equivalents of Fractions of an inch.

| Fractions | Decimals | Fractions. | Decimals | Fractions. | Decimals | Fractions. | Decimals |
|---|---|---|---|---|---|---|---|
| 1/64 | ·015625 | 17/64 | ·265625 | 33/64 | ·515625 | 49/64 | ·765625 |
| 1/32 | ·03125 | 9/32 | ·28125 | 17/32 | ·53125 | 25/32 | ·78125 |
| 3/64 | ·046875 | 19/64 | ·296875 | 35/64 | ·546875 | 51/64 | ·796875 |
| 1/16 | ·0625 | 5/16 | ·3125 | 9/16 | ·5625 | 13/16 | ·8125 |
| 5/64 | ·078125 | 21/64 | ·328125 | 37/64 | ·578125 | 53/64 | ·828125 |
| 3/32 | ·09375 | 11/32 | ·34375 | 19/32 | ·59375 | 27/32 | ·84375 |
| 7/64 | ·109375 | 23/64 | ·359375 | 39/64 | ·609375 | 55/64 | ·859375 |
| ⅛ | ·125 | ⅜ | ·375 | ⅝ | ·625 | ⅞ | ·875 |
| 9/64 | ·140625 | 25/64 | ·390625 | 41/64 | ·640625 | 57/64 | ·890625 |
| 5/32 | ·15625 | 13/32 | ·40625 | 21/32 | ·65625 | 29/32 | ·90625 |
| 11/64 | ·171875 | 27/64 | ·421875 | 43/64 | ·671875 | 59/64 | ·921875 |
| 3/16 | ·1875 | 7/16 | ·4375 | 11/16 | ·6875 | 15/16 | ·9375 |
| 13/64 | ·203125 | 29/64 | ·453125 | 45/64 | ·703125 | 61/64 | ·953125 |
| 7/32 | ·21875 | 15/32 | ·46875 | 23/32 | ·71875 | 31/32 | ·96875 |
| 15/64 | ·234375 | 31/64 | ·484375 | 47/64 | ·734375 | 63/64 | ·984375 |
| ¼ | ·25 | ½ | ·5 | ¾ | ·75 | 1 | 1 |

## Table CLXXXIV.—Metrical equivalents of Fractions of an inch, &c.

| Fractions of an inch. | Millimetres. | Fractions of an inch. | Millimetres. | Inches. | Millimetres. | Inches. | Millimetres. |
|---|---|---|---|---|---|---|---|
| 1/32 | 0·7937 | 17/32 | 13·4935 | 1 | 25·3995 | 17 | 431·7922 |
| 1/16 | 1·5875 | 9/16 | 14·2872 | 2 | 50·7991 | 18 | 457·1917 |
| 3/32 | 2·3812 | 19/32 | 15·0810 | 3 | 76·1986 | 19 | 482·5913 |
| ⅛ | 3·1749 | ⅝ | 15·8747 | 4 | 101·5982 | 20 | 507·9908 |
| 5/32 | 3·9687 | 21/32 | 16·6684 | 5 | 126·9977 | 21 | 533·3904 |
| 3/16 | 4·7624 | 11/16 | 17·4622 | 6 | 152·3972 | 22 | 558·7899 |
| 7/32 | 5·5561 | 23/32 | 18·2559 | 7 | 177·7968 | 23 | 584·1894 |
| ¼ | 6·3499 | ¾ | 19·0496 | 8 | 203·1963 | 24 | 609·5890 |
| 9/32 | 7·1436 | 25/32 | 19·8434 | 9 | 228·5959 | 25 | 634·9885 |
| 5/16 | 7·9374 | 13/16 | 20·6371 | 10 | 253·9954 | 26 | 660·3881 |
| 11/32 | 8·7311 | 27/32 | 21·4309 | 11 | 279·3950 | 27 | 685·7876 |
| ⅜ | 9·5248 | ⅞ | 22·2246 | 12 | 304·7945 | 28 | 711·1872 |
| 13/32 | 10·3186 | 29/32 | 23·0183 | 13 | 330·1940 | 29 | 736·5867 |
| 7/16 | 11·1123 | 15/16 | 23·8121 | 14 | 355·5936 | 30 | 761·9862 |
| 15/32 | 11·9060 | 31/32 | 24·6058 | 15 | 380·9931 | 31 | 787·3858 |
| ½ | 12·6998 |  |  | 16 | 406·3926 | 32 | 812·7853 |

## Table CLXXXV.—Square Feet and Square Metres.

| Square Feet. | Square Metres. | Square Feet. | Square Metres. | Square Feet. | Square Metres. | Square Feet. | Square Metres. |
|---|---|---|---|---|---|---|---|
| 1 | ·0929 | 26 | 2·4154 | 51 | 4·7379 | 76 | 7·0604 |
| 2 | ·1858 | 27 | 2·5083 | 52 | 4·8308 | 77 | 7·1533 |
| 3 | ·2787 | 28 | 2·6012 | 53 | 4·9237 | 78 | 7·2462 |
| 4 | ·3716 | 29 | 2·6941 | 54 | 5·0166 | 79 | 7·3391 |
| 5 | ·4645 | 30 | 2·7870 | 55 | 5·1095 | 80 | 7·4320 |
| 6 | ·5574 | 31 | 2·8799 | 56 | 5·2024 | 81 | 7·5249 |
| 7 | ·6503 | 32 | 2·9728 | 57 | 5·2953 | 82 | 7·6178 |
| 8 | ·7432 | 33 | 3·0657 | 58 | 5·3882 | 83 | 7·7107 |
| 9 | ·8361 | 34 | 3·1586 | 59 | 5·4811 | 84 | 7·8036 |
| 10 | ·9290 | 35 | 3·2515 | 60 | 5·5740 | 85 | 7·8965 |
| 11 | 1·0219 | 36 | 3·3444 | 61 | 5·6669 | 86 | 7·9894 |
| 12 | 1·1148 | 37 | 3·4373 | 62 | 5·7598 | 87 | 8·0823 |
| 13 | 1·2077 | 38 | 3·5302 | 63 | 5·8527 | 88 | 8·1752 |
| 14 | 1·3006 | 39 | 3·6231 | 64 | 5·9456 | 89 | 8·2681 |
| 15 | 1·3935 | 40 | 3·7160 | 65 | 6·0385 | 90 | 8·3610 |
| 16 | 1·4864 | 41 | 3·8089 | 66 | 6·1314 | 91 | 8·4539 |
| 17 | 1·5793 | 42 | 3·9018 | 67 | 6·2243 | 92 | 8·5468 |
| 18 | 1·6722 | 43 | 3·9947 | 68 | 6·3172 | 93 | 8·6397 |
| 19 | 1·7651 | 44 | 4·0876 | 69 | 6·4101 | 94 | 8·7326 |
| 20 | 1·8580 | 45 | 4·1805 | 70 | 6·5030 | 95 | 8·8255 |
| 21 | 1·9509 | 46 | 4·2734 | 71 | 6·5969 | 96 | 8·9184 |
| 22 | 2·0438 | 47 | 4·3663 | 72 | 6·6888 | 97 | 9·0113 |
| 23 | 2·1367 | 48 | 4·4592 | 73 | 6·7817 | 98 | 9·1042 |
| 24 | 2·2296 | 49 | 4·5521 | 74 | 6·8746 | 99 | 9·1971 |
| 25 | 2·3225 | 50 | 4·6450 | 75 | 6·9675 | 100 | 9·2900 |

The above Table can, of course, be used for hundreds and thousands of feet, or for hundredths and thousandths of feet, by altering the position of the decimal point: *e.g* ,—50 square feet = 4·645 square metres, and 5000 square feet = 464·5 square metres; also ·5 square foot = ·04645 square metre.

## Table CLXXXVI.—Square Metres and Square Feet.

| Square Metres. | Square Feet. | Square Metres. | Square Feet. | Square Metres. | Square Feet. | Square Metres. | Square Feet. |
|---|---|---|---|---|---|---|---|
| 1 | 10·764 | 26 | 279·872 | 51 | 548·979 | 76 | 818·087 |
| 2 | 21·529 | 27 | 290·636 | 52 | 559·744 | 77 | 828·851 |
| 3 | 32·293 | 28 | 301·400 | 53 | 570·508 | 78 | 839·615 |
| 4 | 43·057 | 29 | 312·165 | 54 | 581·272 | 79 | 850·380 |
| 5 | 53·822 | 30 | 322·929 | 55 | 592·036 | 80 | 861·144 |
| 6 | 64·586 | 31 | 333·693 | 56 | 602·801 | 81 | 871·908 |
| 7 | 75·350 | 32 | 344·458 | 57 | 613·565 | 82 | 882·673 |
| 8 | 86·114 | 33 | 355·222 | 58 | 624·329 | 83 | 893·437 |
| 9 | 96·879 | 34 | 365·986 | 59 | 635·094 | 84 | 904·201 |
| 10 | 107·643 | 35 | 376·750 | 60 | 645·858 | 85 | 914·966 |
| 11 | 118·407 | 36 | 387·515 | 61 | 656·622 | 86 | 925·730 |
| 12 | 129·172 | 37 | 398·279 | 62 | 667·387 | 87 | 936·494 |
| 13 | 139·936 | 38 | 409·043 | 63 | 678·151 | 88 | 947·258 |
| 14 | 150·700 | 39 | 419·808 | 64 | 688·915 | 89 | 958·023 |
| 15 | 161·464 | 40 | 430·572 | 65 | 699·680 | 90 | 968·787 |
| 16 | 172·229 | 41 | 441·336 | 66 | 710·444 | 91 | 979·551 |
| 17 | 182·993 | 42 | 452·101 | 67 | 721·208 | 92 | 990·316 |
| 18 | 193·757 | 43 | 462·865 | 68 | 731·972 | 93 | 1001·080 |
| 19 | 204·522 | 44 | 473·629 | 69 | 742·737 | 94 | 1011·844 |
| 20 | 215·286 | 45 | 484·394 | 70 | 753·501 | 95 | 1022·608 |
| 21 | 226·050 | 46 | 495·158 | 71 | 764·265 | 96 | 1033·373 |
| 22 | 236·815 | 47 | 505·922 | 72 | 775·030 | 97 | 1044·137 |
| 23 | 247·579 | 48 | 516·686 | 73 | 785·794 | 98 | 1054·901 |
| 24 | 258·343 | 49 | 527·451 | 74 | 796·558 | 99 | 1065·666 |
| 25 | 269·108 | 50 | 538·215 | 75 | 807·322 | 100 | 1076·430 |

The above Table can, of course, be used for hundreds and thousands of metres, or for hundredths and thousandths of metres, by altering the position of the decimal point: *e.g.*,—50 square metres = 538·215 square feet, and 5000 square metres = 53821·5 square feet; also ·5 square metre = 5·38215 square feet.

## Table CLXXXVII.—English Weights and Metric Equivalents

| Lbs. | Kilogrammes. | Lbs. | Kilogrammes. | Lbs. | Kilogrammes. |
|---|---|---|---|---|---|
| 1 | ·4536 | 42 | 19·0509 | 83 | 37·6482 |
| 2 | ·9072 | 43 | 19·5045 | 84 | 38·1018 |
| 3 | 1·3608 | 44 | 19·9581 | 85 | 38·5554 |
| 4 | 1·8144 | 45 | 20·4117 | 86 | 39·0089 |
| 5 | 2·2680 | 46 | 20·8653 | 87 | 39·4625 |
| 6 | 2·7216 | 47 | 21·3189 | 88 | 39·9161 |
| 7 | 3·1752 | 48 | 21·7724 | 89 | 40·3697 |
| 8 | 3·6287 | 49 | 22·2260 | 90 | 40·8233 |
| 9 | 4·0823 | 50 | 22·6796 | 91 | 41·2769 |
| 10 | 4·5359 | 51 | 23·1332 | 92 | 41·7305 |
| 11 | 4·9895 | 52 | 23·5868 | 93 | 42·1841 |
| 12 | 5·4431 | 53 | 24·0404 | 94 | 42·6377 |
| 13 | 5·8967 | 54 | 24·4940 | 95 | 43·0913 |
| 14 | 6·3503 | 55 | 24·9476 | 96 | 43·5449 |
| 15 | 6·8039 | 56 | 25·4012 | 97 | 43·9985 |
| 16 | 7·2575 | 57 | 25·8548 | 98 | 44·4521 |
| 17 | 7·7111 | 58 | 26·3084 | 99 | 44·9057 |
| 18 | 8·1647 | 59 | 26·7619 | 100 | 45·3593 |
| 19 | 8·6182 | 60 | 27·2155 | 101 | 45·8128 |
| 20 | 9·0718 | 61 | 27·6691 | 102 | 46·2664 |
| 21 | 9·5254 | 62 | 28·1227 | 103 | 46·7200 |
| 22 | 9·9790 | 63 | 28·5763 | 104 | 47·1736 |
| 23 | 10·4326 | 64 | 29·0299 | 105 | 47·6272 |
| 24 | 10·8862 | 65 | 29·4835 | 106 | 48·0808 |
| 25 | 11·3398 | 66 | 29·9371 | 107 | 48·5344 |
| 26 | 11·7934 | 67 | 30·3907 | 108 | 48·9880 |
| 27 | 12·2470 | 68 | 30·8443 | 109 | 49·4416 |
| 28 | 12·7006 | 69 | 31·2979 | 110 | 49·8952 |
| 29 | 13·1542 | 70 | 31·7515 | 111 | 50·3488 |
| 30 | 13·6078 | 71 | 32·3051 | 112 | 50·8024 |
| 31 | 14·0614 | 72 | 32·6587 | 200 | 90·7185 |
| 32 | 14·5149 | 73 | 33·1123 | 300 | 136·0778 |
| 33 | 14·9685 | 74 | 33·5658 | 400 | 181·4370 |
| 34 | 15·4221 | 75 | 34·0194 | 500 | 226·7963 |
| 35 | 15·8757 | 76 | 34·4730 | 600 | 272·1556 |
| 36 | 16·3292 | 77 | 34·9266 | 700 | 317·5148 |
| 37 | 16·7293 | 78 | 35·3802 | 800 | 362·8741 |
| 38 | 17·2365 | 79 | 35·8338 | 900 | 408·2334 |
| 39 | 17·6901 | 80 | 36·2874 | 1000 | 453·5926 |
| 40 | 18·1437 | 81 | 36·7410 | 2000 | 907·1853 |
| 41 | 18·5973 | 82 | 37·1946 | 2240 | 1016·0475 |

27 lbs. = 12·247 kilos. : 2·7 lbs. = 1·2247 kilos. : 2700 lbs. = 1224·7 kilos., &c.

## Table CLXXXVIIa.—Metric Weights and English Equivalents.

| Kilo-grams. | Lbs. | Kilo-grams. | Lbs. | Kilo-grams. | Lbs. |
|---|---|---|---|---|---|
| 1 | 2·2046 | 38 | 83·7756 | 75 | 165·3466 |
| 2 | 4·4092 | 39 | 85·9802 | 76 | 167·5512 |
| 3 | 6·6139 | 40 | 88·1848 | 77 | 169·7559 |
| 4 | 8·8185 | 41 | 90·3895 | 78 | 171·9605 |
| 5 | 11·0231 | 42 | 92·5941 | 79 | 174·1651 |
| 6 | 13·2277 | 43 | 94·7987 | 80 | 176·3697 |
| 7 | 15·4324 | 44 | 97·0034 | 81 | 178·5743 |
| 8 | 17·6370 | 45 | 99·2079 | 82 | 180·7789 |
| 9 | 19·8416 | 46 | 101·4126 | 83 | 182·9836 |
| 10 | 22·0462 | 47 | 103·6172 | 84 | 185·1882 |
| 11 | 24·2508 | 48 | 105·8218 | 85 | 187·3928 |
| 12 | 26·4554 | 49 | 108·0264 | 86 | 189·5974 |
| 13 | 28·6601 | 50 | 110·2311 | 87 | 191·8020 |
| 14 | 30·8647 | 51 | 112·4357 | 88 | 194·0067 |
| 15 | 33·0693 | 52 | 114·6403 | 89 | 196·2113 |
| 16 | 35·2739 | 53 | 116·8499 | 90 | 198·4159 |
| 17 | 37·4786 | 54 | 119·0495 | 91 | 200·6205 |
| 18 | 39·6832 | 55 | 121·2542 | 92 | 202·8251 |
| 19 | 41·8878 | 56 | 123·4588 | 93 | 205·0298 |
| 20 | 44·0924 | 57 | 125·6634 | 94 | 207·2344 |
| 21 | 46·2970 | 58 | 127·8680 | 95 | 209·4390 |
| 22 | 48·5017 | 59 | 130·6727 | 96 | 211·6431 |
| 23 | 50·7063 | 60 | 132·2773 | 97 | 213·8482 |
| 24 | 52·9109 | 61 | 134·4819 | 98 | 216·0529 |
| 25 | 55·1155 | 62 | 136·6865 | 99 | 218·2575 |
| 26 | 57·3202 | 63 | 138·8911 | 100 | 220·4621 |
| 27 | 59·5248 | 64 | 141·0958 | 200 | 440·9243 |
| 28 | 61·7294 | 65 | 143·3004 | 300 | 661·3864 |
| 29 | 63·9340 | 66 | 145·5050 | 400 | 881·8485 |
| 30 | 66·1386 | 67 | 147·7096 | 500 | 1102·3106 |
| 31 | 68·3433 | 68 | 149·9142 | 600 | 1322·7728 |
| 32 | 70·5479 | 69 | 152·1189 | 700 | 1543·2349 |
| 33 | 72·7525 | 70 | 154·3235 | 800 | 1763·6970 |
| 34 | 74·9571 | 71 | 156·5281 | 900 | 1984·1591 |
| 35 | 77·1617 | 72 | 158·7327 | 1000 | 2204·6213 |
| 36 | 79·3664 | 73 | 160·9374 | 1016 | 2239·8952 |
| 37 | 81·5709 | 74 | 163·1419 | | |

**58 kilos. = 127·868 lbs. ; 5·8 kilos. = 12·7868 lbs. ; 5800 kilos. = 12786·8 lbs., &c.**

## Table CLXXXVIII.—Pounds per square inch and Kilogrammes per square centimetre.

| Lbs. per sq. inch. | Kilos. per sq. cm. | Lbs. per sq. inch. | Kilos. per sq. cm. | Lbs. per sq. inch. | Kilos. per sq. cm. | Lbs. per sq. inch. | Kilos. per sq. cm. | Lbs. per sq. inch. | Kilos. per sq. cm. |
|---|---|---|---|---|---|---|---|---|---|
| 1 | ·0703 | 35 | 2·460 | 69 | 4·850 | 103 | 7·241 | 137 | 9·632 |
| 2 | ·1406 | 36 | 2·530 | 70 | 4·921 | 104 | 7·312 | 138 | 9·702 |
| 3 | ·2109 | 37 | 2·601 | 71 | 4·991 | 105 | 7·382 | 139 | 9·772 |
| 4 | ·2812 | 38 | 2·671 | 72 | 5·061 | 106 | 7·452 | 140 | 9·843 |
| 5 | ·3515 | 39 | 2·741 | 73 | 5·131 | 107 | 7·522 | 141 | 9·913 |
| 6 | ·4218 | 40 | 2·812 | 74 | 5·202 | 108 | 7·593 | 142 | 9·983 |
| 7 | ·4921 | 41 | 2·882 | 75 | 5·272 | 109 | 7·663 | 143 | 10·054 |
| 8 | ·5624 | 42 | 2·952 | 76 | 5·342 | 110 | 7·733 | 144 | 10·124 |
| 9 | ·6327 | 43 | 3·022 | 77 | 5·413 | 111 | 7·804 | 145 | 10·194 |
| 10 | ·7030 | 44 | 3·093 | 78 | 5·483 | 112 | 7·874 | 146 | 10·264 |
| 11 | ·7733 | 45 | 3·163 | 79 | 5·553 | 113 | 7·944 | 147 | 10·335 |
| 12 | ·8436 | 46 | 3·233 | 80 | 5·624 | 114 | 8·015 | 148 | 10·405 |
| 13 | ·9140 | 47 | 3·304 | 81 | 5·694 | 115 | 8·085 | 149 | 10·475 |
| 14 | ·9843 | 48 | 3·374 | 82 | 5·764 | 116 | 8·155 | 150 | 10·546 |
| 15 | 1·0546 | 49 | 3·444 | 83 | 5·834 | 117 | 8·226 | 155 | 10·897 |
| 16 | 1·1248 | 50 | 3·515 | 84 | 5·905 | 118 | 8·296 | 160 | 11·249 |
| 17 | 1·1952 | 51 | 3·585 | 85 | 5·975 | 119 | 8·366 | 165 | 11·600 |
| 18 | 1·265 | 52 | 3·655 | 86 | 6·045 | 120 | 8·436 | 170 | 11·952 |
| 19 | 1·335 | 53 | 3·725 | 87 | 6·116 | 121 | 8·507 | 175 | 12·303 |
| 20 | 1·406 | 54 | 3·796 | 88 | 6·186 | 122 | 8·577 | 180 | 12·655 |
| 21 | 1·476 | 55 | 3·866 | 89 | 6·256 | 123 | 8·647 | 185 | 13·006 |
| 22 | 1·546 | 56 | 3·936 | 90 | 6·327 | 124 | 8·718 | 190 | 13·358 |
| 23 | 1·616 | 57 | 4·007 | 91 | 6·397 | 125 | 8·788 | 195 | 13·710 |
| 24 | 1·687 | 58 | 4·077 | 92 | 6·467 | 126 | 8·858 | 200 | 14·061 |
| 25 | 1·757 | 59 | 4·147 | 93 | 6·537 | 127 | 8·929 | 210 | 14·76 |
| 26 | 1·827 | 60 | 4·218 | 94 | 6·608 | 128 | 8·999 | 220 | 15·46 |
| 27 | 1·898 | 61 | 4·288 | 95 | 6·678 | 129 | 9·069 | 230 | 16·16 |
| 28 | 1·968 | 62 | 4·358 | 96 | 6·748 | 130 | 9·140 | 240 | 16·87 |
| 29 | 2·038 | 63 | 4·428 | 97 | 6·819 | 131 | 9·210 | 250 | 17·57 |
| 30 | 2·109 | 64 | 4·499 | 98 | 6·889 | 132 | 9·280 | 260 | 18·27 |
| 31 | 2·179 | 65 | 4·569 | 99 | 6·959 | 133 | 9·350 | 270 | 18·98 |
| 32 | 2·249 | 66 | 4·639 | 100 | 7·030 | 134 | 9·421 | 280 | 19·68 |
| 33 | 2·319 | 67 | 4·710 | 101 | 7·101 | 135 | 9·491 | 290 | 20·38 |
| 34 | 2·390 | 68 | 4·780 | 102 | 7·171 | 136 | 9·561 | 300 | 21·09 |

## Table CLXXXIX.—Kilogrammes per square centimetre and Pounds per square inch.

| Kilos. per sq. cm. | Lbs. per square inch. | Kilos. per sq. cm. | Lbs. per square inch. | Kilos. per sq. cm. | Lbs. per square inch. | Kilos. per sq. cm. | Lbs. per square inch. |
|---|---|---|---|---|---|---|---|
| ·1 | 1·422 | 3·1 | 44·091 | 6·1 | 86·761 | 9·1 | 129·431 |
| ·2 | 2·844 | 3·2 | 45·514 | 6·2 | 88·183 | 9·2 | 130·853 |
| ·3 | 4·266 | 3·3 | 46·936 | 6·3 | 89·606 | 9·3 | 132·275 |
| ·4 | 5·689 | 3·4 | 48·358 | 6·4 | 91·028 | 9·4 | 133·698 |
| ·5 | 7·111 | 3·5 | 49·781 | 6·5 | 92·450 | 9·5 | 135·120 |
| ·6 | 8·533 | 3·6 | 51·203 | 6·6 | 93·873 | 9·6 | 136·542 |
| ·7 | 9·956 | 3·7 | 52·625 | 6·7 | 95·295 | 9·7 | 137·965 |
| ·8 | 11·378 | 3·8 | 54·048 | 6·8 | 96·717 | 9·8 | 139·387 |
| ·9 | 12·800 | 3·9 | 55·470 | 6·9 | 98·140 | 9·9 | 140·809 |
| 1·0 | 14·223 | 4·0 | 56·892 | 7·0 | 99·562 | 10·0 | 142·232 |
| 1·1 | 15·645 | 4·1 | 58·315 | 7·1 | 100·984 | 10·5 | 149·343 |
| 1·2 | 17·067 | 4·2 | 59·737 | 7·2 | 102·407 | 11·0 | 156·455 |
| 1·3 | 18·490 | 4·3 | 61·159 | 7·3 | 103·829 | 11·5 | 163·566 |
| 1·4 | 19·912 | 4·4 | 62·582 | 7·4 | 105·251 | 12·0 | 170·678 |
| 1·5 | 21·334 | 4·5 | 64·004 | 7·5 | 106·674 | 12·5 | 177·790 |
| 1·6 | 22·757 | 4·6 | 65·426 | 7·6 | 108·096 | 13·0 | 184·901 |
| 1·7 | 24·179 | 4·7 | 66·849 | 7·7 | 109·518 | 13·5 | 192·013 |
| 1·8 | 25·601 | 4·8 | 68·271 | 7·8 | 110·940 | 14·0 | 199·124 |
| 1·9 | 27·024 | 4·9 | 69·693 | 7·9 | 112·363 | 14·5 | 206·236 |
| 2·0 | 28·446 | 5·0 | 71·116 | 8·0 | 113·785 | 15·0 | 213·348 |
| 2·1 | 29·868 | 5·1 | 72·538 | 8·1 | 115·207 | 15·5 | 220·459 |
| 2·2 | 31·291 | 5·2 | 73·960 | 8·2 | 116·630 | 16·0 | 227·571 |
| 2·3 | 32·713 | 5·3 | 75·382 | 8·3 | 118·052 | 16·5 | 234·682 |
| 2·4 | 34·135 | 5·4 | 76·805 | 8·4 | 119·474 | 17·0 | 241·794 |
| 2·5 | 35·558 | 5·5 | 78·227 | 8·5 | 120·897 | 17·5 | 248·906 |
| 2·6 | 36·980 | 5·6 | 79·649 | 8·6 | 122·319 | 18·0 | 256·017 |
| 2·7 | 38·402 | 5·7 | 81·072 | 8·7 | 123·741 | 18·5 | 263·129 |
| 2·8 | 39·824 | 5·8 | 82·494 | 8·8 | 125·164 | 19·0 | 270·240 |
| 2·9 | 41·247 | 5·9 | 83·916 | 8·9 | 126·586 | 19·5 | 277·352 |
| 3·0 | 42·669 | 6·0 | 85·339 | 9·0 | 128·008 | 20·0 | 284·464 |

## Table CXC.—Kilogrammes per square millimetre (or centimetre) and Tons per square inch.*

| Kilos. per mm.$^2$ | Tons per sq. in. | Kilos. per mm.$^2$ | Tons per sq. in. | Kilos. per mm.$^2$ | Tons per sq. in. | Kilos. per mm.$^2$ | Tons per sq. in. |
|---|---|---|---|---|---|---|---|
| ·5 | ·318 | 17·5 | 11·112 | 34·5 | 21·908 | 51·5 | 32·702 |
| 1·0 | ·635 | 18·0 | 11·430 | 35·0 | 22·225 | 52·0 | 33·020 |
| 1·5 | ·952 | 18·5 | 11·748 | 35·5 | 22·542 | 52·5 | 33·338 |
| 2·0 | 1·270 | 19·0 | 12·065 | 36·0 | 22·860 | 53·0 | 33·655 |
| 2·5 | 1·588 | 19·5 | 12·382 | 36·5 | 23·178 | 53·5 | 33·972 |
| 3·0 | 1·905 | 20·0 | 12·700 | 37·0 | 23·495 | 54·0 | 34·290 |
| 3·5 | 2·222 | 20·5 | 13·018 | 37·5 | 23·812 | 54·5 | 34·608 |
| 4·0 | 2·540 | 21·0 | 13·335 | 38·0 | 24·130 | 55·0 | 34·925 |
| 4·5 | 2·858 | 21·5 | 13·652 | 38·5 | 24·448 | 55·5 | 35·232 |
| 5·0 | 3·175 | 22·0 | 13·970 | 39·0 | 24·765 | 56·0 | 35·560 |
| 5·5 | 3·492 | 22·5 | 14·288 | 39·5 | 25·082 | 56·5 | 35·878 |
| 6·0 | 3·810 | 23·0 | 14·605 | 40·0 | 25·400 | 57·0 | 36·185 |
| 6·5 | 4·128 | 23·5 | 14·922 | 40·5 | 25·718 | 57·5 | 36·502 |
| 7·0 | 4·445 | 24·0 | 15·240 | 41·0 | 26·035 | 58·0 | 36·830 |
| 7·5 | 4·762 | 24·5 | 15·558 | 41·5 | 26·352 | 58·5 | 37·148 |
| 8·0 | 5·080 | 25·0 | 15·875 | 42·0 | 26·670 | 59·0 | 37·465 |
| 8·5 | 5·398 | 25·5 | 16·192 | 42·5 | 26·988 | 59·5 | 37·472 |
| 9·0 | 5·715 | 26·0 | 16·510 | 43·0 | 27·305 | 60·0 | 38·100 |
| 9·5 | 6·032 | 26·5 | 16·828 | 43·5 | 27·622 | 60·5 | 38·418 |
| 10·0 | 6·350 | 27·0 | 17·145 | 44·0 | 27·940 | 61·0 | 38·735 |
| 10·5 | 6·668 | 27·5 | 17·462 | 44·5 | 28·258 | 61·5 | 39·052 |
| 11·0 | 6·985 | 28·0 | 17·780 | 45·0 | 28·575 | 62·0 | 39·370 |
| 11·5 | 7·302 | 28·5 | 18·098 | 45·5 | 28·892 | 62·5 | 39·688 |
| 12·0 | 7·620 | 29·0 | 18·415 | 46·0 | 29·210 | 63·0 | 40·005 |
| 12·5 | 7·938 | 29·5 | 18·732 | 46·5 | 29·528 | 63·5 | 40·322 |
| 13·0 | 8·255 | 30·0 | 19·050 | 47·0 | 29·845 | 64·0 | 40·640 |
| 13·5 | 8·572 | 30·5 | 19·368 | 47·5 | 30·162 | 64·5 | 40·958 |
| 14·0 | 8·890 | 31·0 | 19·685 | 48·0 | 30·480 | 65·0 | 41·275 |
| 14·5 | 9·208 | 31·5 | 20·002 | 48·5 | 30·798 | 65·5 | 41·592 |
| 15·0 | 9·525 | 32·0 | 20·320 | 49·0 | 31·115 | 66·0 | 41·910 |
| 15·5 | 9·842 | 32·5 | 20·638 | 49·5 | 31·432 | 66·5 | 42·238 |
| 16·0 | 10·160 | 33·0 | 20·955 | 50·0 | 31·750 | 67·0 | 42·545 |
| 16·5 | 10·478 | 33·5 | 21·272 | 50·5 | 32·068 | 67·5 | 42·852 |
| 17·0 | 10·795 | 34·0 | 21·590 | 51·0 | 32·385 | 68·0 | 43·170 |

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* **1 kilo. per sq. millimetre = 100 kilos. per sq. centimetre, etc.**

## Table CXCI.—Areas of Segments of Circles.

To find the area of any segment of a circle,—Divide the versed sine or height of the segment (V) by the diameter of the circle of which it is a part (D), and multiply the square of the diameter by the value of $x$ (see Table) corresponding to the value of $\frac{V}{D}$ obtained; that is,—

$$\text{Area of Segment} = \text{Diameter}^2 \times x.$$

| $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ |
|---|---|---|---|---|---|---|---|
| ·001 | ·000042 | ·038 | ·009763 | ·075 | ·026761 | ·112 | ·048262 |
| ·002 | ·000119 | ·039 | ·010148 | ·076 | ·027289 | ·113 | ·048894 |
| ·003 | ·000219 | ·040 | ·010537 | ·077 | ·027821 | ·114 | ·049528 |
| ·004 | ·000337 | ·041 | ·010931 | ·078 | ·028356 | ·115 | ·050165 |
| ·005 | ·000470 | ·042 | ·011330 | ·079 | ·028894 | ·116 | ·050804 |
| ·006 | ·000618 | ·043 | ·011734 | ·080 | ·029435 | ·117 | ·051446 |
| ·007 | ·000779 | ·044 | ·012142 | ·081 | ·029979 | ·118 | ·052090 |
| ·008 | ·000951 | ·045 | ·012554 | ·082 | ·030526 | ·119 | ·052736 |
| ·009 | ·001135 | ·046 | ·012971 | ·083 | ·031076 | ·120 | ·053385 |
| ·010 | ·001329 | ·047 | ·013392 | ·084 | ·031629 | ·121 | ·054036 |
| ·011 | ·001533 | ·048 | ·013818 | ·085 | ·032186 | ·122 | ·054689 |
| ·012 | ·001746 | ·049 | ·014247 | ·086 | ·032745 | ·123 | ·055345 |
| ·013 | ·001968 | ·050 | ·014681 | ·087 | ·033307 | ·124 | ·056003 |
| ·014 | ·002199 | ·051 | ·015119 | ·088 | ·033872 | ·125 | ·056663 |
| ·015 | ·002438 | ·052 | ·015561 | ·089 | ·034441 | ·126 | ·057326 |
| ·016 | ·002685 | ·053 | ·016007 | ·090 | ·035011 | ·127 | ·057991 |
| ·017 | ·002940 | ·054 | ·016457 | ·091 | ·035585 | ·128 | ·058658 |
| ·018 | ·003202 | ·055 | ·016911 | ·092 | ·036162 | ·129 | ·059327 |
| ·019 | ·003471 | ·056 | ·017369 | ·093 | ·036741 | ·130 | ·059999 |
| ·020 | ·003748 | ·057 | ·017831 | ·094 | ·037323 | ·131 | ·060672 |
| ·021 | ·004031 | ·058 | ·018296 | ·095 | ·037909 | ·132 | ·061348 |
| ·022 | ·004322 | ·059 | ·018766 | ·096 | ·038496 | ·133 | ·062026 |
| ·023 | ·004618 | ·060 | ·019239 | ·097 | ·039087 | ·134 | ·062707 |
| ·024 | ·004921 | ·061 | ·019716 | ·098 | ·039680 | ·135 | ·063389 |
| ·025 | ·005230 | ·062 | ·020196 | ·099 | ·040276 | ·136 | ·064074 |
| ·026 | ·005546 | ·063 | ·020680 | ·100 | ·040875 | ·137 | ·064760 |
| ·027 | ·005867 | ·064 | ·021168 | ·101 | ·041476 | ·138 | ·065449 |
| ·028 | ·006194 | ·065 | ·021659 | ·102 | ·042080 | ·139 | ·066140 |
| ·029 | ·006527 | ·066 | ·022154 | ·103 | ·042687 | ·140 | ·066833 |
| ·030 | ·006865 | ·067 | ·022652 | ·104 | ·043296 | ·141 | ·067528 |
| ·031 | ·007209 | ·068 | ·023154 | ·105 | ·043908 | ·142 | ·068225 |
| ·032 | ·007558 | ·069 | ·023659 | ·106 | ·044522 | ·143 | ·068924 |
| ·033 | ·007913 | ·070 | ·024168 | ·107 | ·045139 | ·144 | ·069625 |
| ·034 | ·008273 | ·071 | ·024680 | ·108 | ·045759 | ·145 | ·070328 |
| ·035 | ·008638 | ·072 | ·025195 | ·109 | ·046381 | ·146 | ·071033 |
| ·036 | ·009008 | ·073 | ·025714 | ·110 | ·047005 | ·147 | ·071741 |
| ·037 | ·009383 | ·074 | ·026236 | ·111 | ·047632 | ·148 | ·072450 |

## Table CXCI.—Areas of Segments of Circles—*continued.*

| $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ |
|---|---|---|---|---|---|---|---|
| ·149 | ·073161 | ·193 | ·106261 | ·237 | ·142387 | ·281 | ·180918 |
| ·150 | ·073874 | ·194 | ·107051 | ·238 | ·143238 | ·282 | ·181817 |
| ·151 | ·074589 | ·195 | ·107842 | ·239 | ·144091 | ·283 | ·182718 |
| ·152 | ·075306 | ·196 | ·108636 | ·240 | ·144944 | ·284 | ·183619 |
| ·153 | ·076026 | ·197 | ·109430 | ·241 | ·145799 | ·285 | ·184521 |
| ·154 | ·076747 | ·198 | ·110226 | ·242 | ·146655 | ·286 | ·185425 |
| ·155 | ·077469 | ·199 | ·111024 | ·243 | ·147512 | ·287 | ·186329 |
| ·156 | ·078194 | ·200 | ·111823 | ·244 | ·148371 | ·288 | ·187234 |
| ·157 | ·078921 | ·201 | ·112624 | ·245 | ·149230 | ·289 | ·188140 |
| ·158 | ·079649 | ·202 | ·113426 | ·246 | ·150091 | ·290 | ·189047 |
| ·159 | ·080380 | ·203 | ·114230 | ·247 | ·150953 | ·291 | ·189955 |
| ·160 | ·081112 | ·204 | ·115035 | ·248 | ·151816 | ·292 | ·190864 |
| ·161 | ·081846 | ·205 | ·115842 | ·249 | ·152680 | ·293 | ·191775 |
| ·162 | ·082582 | ·206 | ·116650 | ·250 | ·153546 | ·294 | ·192684 |
| ·163 | ·083320 | ·207 | ·117460 | ·251 | ·154412 | ·295 | ·193596 |
| ·164 | ·084059 | ·208 | ·118271 | ·252 | ·155280 | ·296 | ·194509 |
| ·165 | ·084801 | ·209 | ·119083 | ·253 | ·156149 | ·297 | ·195422 |
| ·166 | ·085544 | ·210 | ·119897 | ·254 | ·157019 | ·298 | ·196337 |
| ·167 | ·086289 | ·211 | ·120712 | ·255 | ·157890 | ·299 | ·197252 |
| ·168 | ·087036 | ·212 | ·121529 | ·256 | ·158762 | ·300 | ·198168 |
| ·169 | ·087785 | ·213 | ·122347 | ·257 | ·159636 | ·301 | ·199085 |
| ·170 | ·088535 | ·214 | ·123167 | ·258 | ·160510 | ·302 | ·200003 |
| ·171 | ·089287 | ·215 | ·123988 | ·259 | ·161386 | ·303 | ·200922 |
| ·172 | ·090041 | ·216 | ·124810 | ·260 | ·162263 | ·304 | ·201841 |
| ·173 | ·090797 | ·217 | ·125634 | ·261 | ·163140 | ·305 | ·202761 |
| ·174 | ·091554 | ·218 | ·126459 | ·262 | ·164019 | ·306 | ·203683 |
| ·175 | ·092313 | ·219 | ·127285 | ·263 | ·164899 | ·307 | ·204605 |
| ·176 | ·093074 | ·220 | ·128113 | ·264 | ·165780 | ·308 | ·205527 |
| ·177 | ·093836 | ·221 | ·128942 | ·265 | ·166663 | ·309 | ·206451 |
| ·178 | ·094601 | ·222 | ·129773 | ·266 | ·167546 | ·310 | ·207376 |
| ·179 | ·095366 | ·223 | ·130605 | ·267 | ·168430 | ·311 | ·208301 |
| ·180 | ·096134 | ·224 | ·131438 | ·268 | ·169315 | ·312 | ·209227 |
| ·181 | ·096903 | ·225 | ·132272 | ·269 | ·170202 | ·313 | ·210154 |
| ·182 | ·097674 | ·226 | ·133108 | ·270 | ·171089 | ·314 | ·211082 |
| ·183 | ·098447 | ·227 | 133945 | ·271 | ·171978 | ·315 | ·212011 |
| ·184 | ·099221 | ·228 | ·134784 | ·272 | ·172867 | ·316 | ·212940 |
| 185 | ·099997 | ·229 | ·135624 | ·273 | ·173758 | ·317 | ·213871 |
| ·186 | 100774 | ·230 | ·136465 | ·274 | ·174649 | ·318 | ·214802 |
| ·187 | ·101553 | ·231 | ·137307 | ·275 | ·175542 | ·319 | ·215733 |
| ·188 | ·102334 | ·232 | ·138150 | ·276 | ·176435 | ·320 | ·216666 |
| ·189 | ·103116 | ·233 | ·138995 | ·277 | ·177330 | ·321 | ·217599 |
| ·190 | ·103900 | ·234 | ·139841 | ·278 | ·178225 | ·322 | ·218533 |
| ·191 | ·104685 | ·235 | ·140688 | ·279 | ·179122 | ·323 | ·219468 |
| ·192 | ·105472 | ·236 | ·141537 | ·280 | ·180019 | ·324 | ·220404 |

## Table CXCI.—Areas of Segments of Circles—*continued.*

| $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ | $\frac{V}{D}$ | $x$ |
|---|---|---|---|---|---|---|---|
| ·325 | ·221340 | ·369 | ·263213 | ·413 | ·306140 | ·457 | ·349752 |
| ·326 | ·222277 | ·370 | ·264178 | ·414 | ·307125 | ·458 | ·350748 |
| ·327 | ·223215 | ·371 | ·265144 | ·415 | ·308110 | ·459 | ·351745 |
| ·328 | ·224154 | ·372 | ·266111 | ·416 | ·309095 | ·460 | ·352742 |
| ·329 | ·225093 | ·373 | ·267078 | ·417 | ·310081 | ·461 | ·353739 |
| ·330 | ·226033 | ·374 | ·268045 | ·418 | ·311068 | ·462 | ·354736 |
| ·331 | ·226974 | ·375 | ·269013 | ·419 | ·312054 | ·463 | ·355732 |
| ·332 | ·227915 | ·376 | ·269982 | ·420 | ·313041 | ·464 | ·356730 |
| ·333 | ·228858 | ·377 | ·270951 | ·421 | ·314029 | ·465 | ·357727 |
| ·334 | ·229801 | ·378 | ·271920 | ·422 | ·315016 | ·466 | ·358725 |
| ·335 | ·230745 | ·379 | ·272890 | ·423 | ·316004 | ·467 | ·359723 |
| ·336 | ·231689 | ·380 | ·273861 | ·424 | ·316992 | ·468 | ·360721 |
| ·337 | ·232634 | ·381 | ·274832 | ·425 | ·317981 | ·469 | ·361719 |
| ·338 | ·233580 | ·382 | ·275803 | ·426 | ·318970 | ·470 | ·362717 |
| ·339 | ·234526 | ·383 | ·276775 | ·427 | ·319959 | ·471 | ·363715 |
| ·340 | ·235473 | ·384 | ·277748 | ·428 | ·320948 | ·472 | ·364713 |
| ·341 | ·236421 | ·385 | ·278721 | ·429 | ·321938 | ·473 | ·365712 |
| ·342 | ·237369 | ·386 | ·279694 | ·430 | ·322928 | ·474 | ·366710 |
| ·343 | ·238318 | ·387 | ·280668 | ·431 | ·323918 | ·475 | ·367709 |
| ·344 | ·239268 | ·388 | ·281642 | ·432 | ·324909 | ·476 | ·368708 |
| ·345 | ·240218 | ·389 | ·282617 | ·433 | ·325900 | ·477 | ·369707 |
| ·346 | ·241169 | ·390 | ·283592 | ·434 | ·326892 | ·478 | ·370706 |
| ·347 | ·242121 | ·391 | ·284568 | ·435 | ·327882 | ·479 | ·371705 |
| ·348 | ·243074 | ·392 | ·285544 | ·436 | ·328874 | ·480 | ·372704 |
| ·349 | ·244026 | ·393 | ·286521 | ·437 | ·329866 | ·481 | ·373703 |
| ·350 | ·244980 | ·394 | ·287498 | ·438 | ·330858 | ·482 | ·374702 |
| ·351 | ·245934 | ·395 | ·288476 | ·439 | ·331850 | ·483 | ·375702 |
| ·352 | ·246889 | ·396 | ·289453 | ·440 | ·332843 | ·484 | ·376702 |
| ·353 | ·247845 | ·397 | ·290432 | ·441 | ·333836 | ·485 | ·377701 |
| ·354 | ·248801 | ·398 | ·291411 | ·442 | ·334829 | ·486 | ·378701 |
| ·355 | ·249757 | ·399 | ·292390 | ·443 | ·335822 | ·487 | ·379700 |
| ·356 | ·250715 | ·400 | ·293369 | ·444 | ·336816 | ·488 | ·380700 |
| ·357 | ·251673 | ·401 | ·294349 | ·445 | ·337810 | ·489 | ·381699 |
| ·358 | ·252631 | ·402 | ·295330 | ·446 | ·338804 | ·490 | ·382699 |
| ·359 | ·253590 | ·403 | ·296311 | ·447 | ·339798 | ·491 | ·383699 |
| ·360 | ·254550 | ·404 | ·297292 | ·448 | ·340793 | ·492 | ·384699 |
| ·361 | ·255510 | ·405 | ·298273 | ·449 | ·341787 | ·493 | ·385699 |
| ·362 | ·256471 | ·406 | ·299255 | ·450 | ·342782 | ·494 | ·386699 |
| ·363 | ·257433 | ·407 | ·300238 | ·451 | ·343777 | ·495 | ·387699 |
| ·364 | ·258395 | ·408 | ·301220 | ·452 | ·344772 | ·496 | ·388699 |
| ·365 | ·259357 | ·409 | ·302203 | ·453 | ·345768 | ·497 | ·389699 |
| ·366 | ·260320 | ·410 | ·303187 | ·454 | ·346764 | ·498 | ·390699 |
| ·367 | ·261284 | ·411 | ·304171 | ·455 | ·347759 | ·499 | ·391699 |
| ·368 | ·262248 | ·412 | ·305155 | ·456 | ·348755 | ·500 | ·392699 |

## Table CXCII.—Areas of Circles.

| Diameters. | Areas. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 0 | ·0 | ·01227 | ·0491 | ·1104 | ·1963 | ·3068 | ·4418 | ·6013 | 0 |
| 1 | ·78540 | ·99402 | 1·2272 | 1·4849 | 1·7671 | 2·0739 | 2·4053 | 2·7612 | 1 |
| 2 | 3·1416 | 3·5466 | 3·9761 | 4·4301 | 4·9087 | 5·4119 | 5·9396 | 6·4918 | 2 |
| 3 | 7·0686 | 7·6699 | 8·2958 | 8·9462 | 9·6211 | 10·3206 | 11·0447 | 11·7933 | 3 |
| 4 | 12·5664 | 13·3641 | 14·1863 | 15·0330 | 15·9043 | 16·8002 | 17·7206 | 18·6655 | 4 |
| 5 | 19·6350 | 20·6290 | 21·6476 | 22·6907 | 23·7583 | 24·8505 | 25·9673 | 27·1086 | 5 |
| 6 | 28·2744 | 29·4648 | 30·6797 | 31·9191 | 33·1831 | 34·4717 | 35·7848 | 37·1224 | 6 |
| 7 | 38·4846 | 39·8713 | 41·2826 | 42·7184 | 44·1787 | 45·6636 | 47·1731 | 48·7071 | 7 |
| 8 | 50·2656 | 51·8487 | 53·4563 | 55·0884 | 56·7451 | 58·4264 | 60·1322 | 61·8625 | 8 |
| 9 | 63·6174 | 65·3968 | 67·2008 | 69·0293 | 70·8823 | 72·7599 | 74·6621 | 76·5888 | 9 |
| 10 | 78·5400 | 80·5158 | 82·5161 | 84·5409 | 86·5903 | 88·6643 | 90·7628 | 92·8858 | 10 |
| 11 | 95·0334 | 97·2055 | 99·5022 | 101·6234 | 103·8691 | 106·1394 | 108·4343 | 110·7537 | 11 |
| 12 | 113·098 | 115·466 | 117·859 | 120·277 | 122·719 | 125·185 | 127·677 | 130·192 | 12 |
| 13 | 132·733 | 135·297 | 137·887 | 140·501 | 143·139 | 145·802 | 148·490 | 151·202 | 13 |
| 14 | 153·938 | 156·700 | 159·485 | 162·296 | 165·130 | 167·990 | 170·874 | 173·782 | 14 |
| 15 | 176·715 | 179·673 | 182·655 | 185·661 | 188·692 | 191·748 | 194·828 | 197·933 | 15 |
| 16 | 201·062 | 204·216 | 207·395 | 210·598 | 213·825 | 217·077 | 220·354 | 223·655 | 16 |
| 17 | 226·981 | 230·331 | 233·706 | 237·105 | 240·529 | 243·977 | 247·450 | 250·948 | 17 |
| 18 | 254·470 | 258·016 | 261·587 | 265·183 | 268·803 | 272·448 | 276·117 | 279·811 | 18 |
| 19 | 283·529 | 287·272 | 291·040 | 294·832 | 298·648 | 302·489 | 306·355 | 310·245 | 19 |
| 20 | 314·160 | 318·099 | 322·063 | 326·051 | 330·064 | 334·102 | 338·164 | 342·250 | 20 |
| 21 | 346·361 | 350·497 | 354·657 | 358·842 | 363·051 | 367·285 | 371·543 | 375·826 | 21 |
| 22 | 380·134 | 384·466 | 388·822 | 393·203 | 397·609 | 402·038 | 406·494 | 410·973 | 22 |

## Table CXCII.—Areas of Circles—*continued.*

| Diameters. | Areas. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 23 | 415·477 | 420·004 | 424·558 | 429·135 | 433·737 | 438·364 | 443·015 | 447·690 | 23 |
| 24 | 452·390 | 457·115 | 461·864 | 466·638 | 471·436 | 476·259 | 481·107 | 485·979 | 24 |
| 25 | 490·875 | 495·796 | 500·742 | 505·712 | 510·706 | 515·726 | 520·769 | 525·838 | 25 |
| 26 | 530·930 | 536·048 | 541·190 | 546·356 | 551·547 | 556·763 | 562·003 | 567·267 | 26 |
| 27 | 572·557 | 577·870 | 583·209 | 588·571 | 593·959 | 599·371 | 604·807 | 610·268 | 27 |
| 28 | 615·754 | 621·264 | 626·798 | 632·357 | 637·941 | 643·549 | 649·182 | 654·840 | 28 |
| 29 | 660·521 | 666·228 | 671·959 | 677·714 | 683·494 | 689·299 | 695·128 | 700·982 | 29 |
| 30 | 706·860 | 712·763 | 718·690 | 724·642 | 730·618 | 736·619 | 742·645 | 748·695 | 30 |
| 31 | 754·769 | 760·869 | 766·992 | 773·140 | 779·313 | 785·510 | 791·732 | 797·979 | 31 |
| 32 | 804·250 | 810·545 | 816·865 | 823·210 | 829·579 | 835·972 | 842·391 | 848·833 | 32 |
| 33 | 855·301 | 861·792 | 868·309 | 874·850 | 881·415 | 888·005 | 894·620 | 901·259 | 33 |
| 34 | 907·922 | 914·611 | 921·323 | 928·061 | 934·822 | 941·609 | 948·420 | 955·255 | 34 |
| 35 | 962·115 | 969·000 | 975·909 | 982·842 | 989·800 | 996·783 | 1003·790 | 1010·822 | 35 |
| 36 | 1017·878 | 1024·960 | 1032·065 | 1039·195 | 1046·349 | 1053·528 | 1060·732 | 1067·960 | 36 |
| 37 | 1075·213 | 1082·490 | 1089·792 | 1097·118 | 1104·469 | 1111·844 | 1119·244 | 1126·669 | 37 |
| 38 | 1134·118 | 1141·591 | 1149·089 | 1156·612 | 1164·159 | 1171·731 | 1179·327 | 1186·948 | 38 |
| 39 | 1194·593 | 1202·263 | 1209·958 | 1217·677 | 1225·420 | 1233·188 | 1240·981 | 1248·798 | 39 |
| 40 | 1256·64 | 1264·51 | 1272·40 | 1280·31 | 1288·25 | 1296·22 | 1304·21 | 1312·22 | 40 |
| 41 | 1320·26 | 1328·32 | 1336·41 | 1344·52 | 1352·66 | 1360·82 | 1369·00 | 1377·21 | 41 |
| 42 | 1385·45 | 1393·70 | 1401·99 | 1410·30 | 1418·63 | 1426·99 | 1435·37 | 1443·77 | 42 |
| 43 | 1452·20 | 1460·66 | 1469·14 | 1477·64 | 1486·17 | 1494·73 | 1503·30 | 1511·91 | 43 |
| 44 | 1520·53 | 1529·19 | 1537·86 | 1546·56 | 1555·29 | 1564·04 | 1572·81 | 1581·61 | 44 |

## Table CXCII.—Areas of Circles—*continued.*

| Diameters. | Areas. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 45 | 1590·43 | 1599·28 | 1608·16 | 1617·05 | 1625·97 | 1634·92 | 1643·89 | 1652·89 | 45 |
| 46 | 1661·91 | 1670·95 | 1680·02 | 1689·11 | 1698·23 | 1707·37 | 1716·54 | 1725·73 | 46 |
| 47 | 1734·95 | 1744·19 | 1753·45 | 1762·74 | 1772·06 | 1781·40 | 1790·76 | 1800·15 | 47 |
| 48 | 1809·56 | 1819·00 | 1828·46 | 1837·95 | 1847·46 | 1856·99 | 1866·55 | 1876·14 | 48 |
| 49 | 1885·75 | 1895·38 | 1905·04 | 1914·72 | 1924·43 | 1934·16 | 1943·91 | 1953·69 | 49 |
| 50 | 1963·50 | 1973·33 | 1983·18 | 1993·06 | 2002·97 | 2012·89 | 2022·85 | 2032·82 | 50 |
| 51 | 2042·83 | 2052·85 | 2062·90 | 2072·98 | 2083·08 | 2093·20 | 2103·35 | 2113·52 | 51 |
| 52 | 2123·72 | 2133·94 | 2144·19 | 2154·46 | 2164·76 | 2175·08 | 2185·42 | 2195·79 | 52 |
| 53 | 2206·19 | 2216·61 | 2227·05 | 2237·52 | 2248·01 | 2258·53 | 2269·07 | 2279·64 | 53 |
| 54 | 2290·23 | 2300·84 | 2311·48 | 2322·15 | 2332·83 | 2343·55 | 2354·29 | 2365·05 | 54 |
| 55 | 2375·83 | 2386·65 | 2397·48 | 2408·34 | 2419·23 | 2430·14 | 2441·07 | 2452·03 | 55 |
| 56 | 2463·01 | 2474·02 | 2485·05 | 2496·11 | 2507·19 | 2518·30 | 2529·43 | 2540·58 | 56 |
| 57 | 2551·76 | 2562·97 | 2574·20 | 2585·45 | 2596·73 | 2608·03 | 2619·36 | 2630·71 | 57 |
| 58 | 2642·09 | 2653·49 | 2664·91 | 2676·36 | 2687·84 | 2699·33 | 2710·86 | 2722·41 | 58 |
| 59 | 2733·98 | 2745·57 | 2757·20 | 2768·84 | 2780·51 | 2792·21 | 2803·93 | 2815·67 | 59 |
| 60 | 2827·44 | 2839·23 | 2851·05 | 2862·89 | 2874·76 | 2886·65 | 2898·57 | 2910·51 | 60 |
| 61 | 2922·47 | 2934·46 | 2946·48 | 2958·52 | 2970·58 | 2982·67 | 2994·78 | 3006·92 | 61 |
| 62 | 3019·08 | 3031·26 | 3043·47 | 3055·71 | 3067·97 | 3080·25 | 3092·56 | 3104·89 | 62 |
| 63 | 3117·25 | 3129·64 | 3142·04 | 3154·47 | 3166·93 | 3179·41 | 3191·91 | 3204·44 | 63 |
| 64 | 3217·00 | 3229·58 | 3242·18 | 3254·81 | 3267·46 | 3280·14 | 3292·84 | 3305·56 | 64 |
| 65 | 3318·31 | 3331·09 | 3343·89 | 3356·71 | 3369·56 | 3382·44 | 3395·33 | 3408·26 | 65 |
| 66 | 3421·20 | 3434·17 | 3447·17 | 3460·19 | 3473·24 | 3486·30 | 3499·40 | 3512·52 | 66 |

## Table CXCII.—Areas of Circles—*continued.*

| Diameters. | Areas. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 67 | 3525·66 | 3538·83 | 3552·02 | 3565·24 | 3578·48 | 3591·74 | 3605·04 | 3618·35 | 67 |
| 68 | 3631·69 | 3645·05 | 3658·44 | 3671·86 | 3685·29 | 3698·76 | 3712·24 | 3725·75 | 68 |
| 69 | 3739·29 | 3752·85 | 3766·43 | 3780·04 | 3793·68 | 3807·34 | 3821·02 | 3834·73 | 69 |
| 70 | 3848·46 | 3862·22 | 3876·00 | 3889·80 | 3903·63 | 3917·49 | 3931·37 | 3945·27 | 70 |
| 71 | 3959·20 | 3973·15 | 3987·13 | 4001·13 | 4015·16 | 4029·21 | 4043·29 | 4057·39 | 71 |
| 72 | 4071·51 | 4085·66 | 4099·84 | 4114·04 | 4128·26 | 4142·51 | 4156·78 | 4171·08 | 72 |
| 73 | 4185·40 | 4199·74 | 4214·11 | 4228·51 | 4242·93 | 4257·37 | 4271·84 | 4286·83 | 73 |
| 74 | 4300·85 | 4315·39 | 4329·96 | 4344·55 | 4359·17 | 4373·81 | 4388·47 | 4403·16 | 74 |
| 75 | 4417·87 | 4432·61 | 4447·38 | 4462·16 | 4476·98 | 4491·81 | 4506·67 | 4521·56 | 75 |
| 76 | 4536·47 | 4551·41 | 4566·36 | 4581·35 | 4596·36 | 4611·39 | 4626·45 | 4641·58 | 76 |
| 77 | 4656·64 | 4671·77 | 4686·92 | 4702·10 | 4717·31 | 4732·54 | 4747·79 | 4763·07 | 77 |
| 78 | 4778·37 | 4793·70 | 4809·05 | 4824·43 | 4839·83 | 4855·26 | 4870·71 | 4886·18 | 78 |
| 79 | 4901·68 | 4917·21 | 4932·75 | 4948·33 | 4963·92 | 4979·55 | 4995·19 | 5010·86 | 79 |
| 80 | 5026·56 | 5042·28 | 5058·03 | 5073·79 | 5089·59 | 5105·41 | 5121·25 | 5137·12 | 80 |
| 81 | 5153·01 | 5168·93 | 5184·87 | 5200·83 | 5216·82 | 5232·84 | 5248·88 | 5264·94 | 81 |
| 82 | 5281·03 | 5297·14 | 5313·28 | 5329·44 | 5345·63 | 5361·84 | 5378·08 | 5394·34 | 82 |
| 83 | 5410·62 | 5426·93 | 5443·26 | 5459·62 | 5476·01 | 5492·41 | 5508·84 | 5525·30 | 83 |
| 84 | 5541·78 | 5558·29 | 5574·82 | 5591·37 | 5607·95 | 5624·56 | 5641·18 | 5657·84 | 84 |
| 85 | 5674·51 | 5691·22 | 5707·94 | 5724·69 | 5741·47 | 5758·27 | 5775·10 | 5791·94 | 85 |
| 86 | 5808·82 | 5825·72 | 5842·64 | 5859·59 | 5876·56 | 5893·55 | 5910·58 | 5927·62 | 86 |
| 87 | 5944·69 | 5961·79 | 5978·91 | 5996·05 | 6013·22 | 6030·41 | 6047·63 | 6064·87 | 87 |
| 88 | 6082·14 | 6099·43 | 6116·74 | 6134·08 | 6151·45 | 6168·84 | 6186·25 | 6203·69 | 88 |

## Table CXCII.—Areas of Circles—*continued.*

| Diameters. | Areas. | | | | | | | | Diameters |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 89 | 6221·15 | 6238·64 | 6256·15 | 6273·69 | 6291·25 | 6308·84 | 6326·45 | 6344·08 | 89 |
| 90 | 6361·74 | 6379·42 | 6397·13 | 6414·86 | 6432·62 | 6450·40 | 6468·21 | 6486·04 | 90 |
| 91 | 6503·90 | 6521·78 | 6539·68 | 6557·61 | 6575·56 | 6593·54 | 6611·55 | 6629·57 | 91 |
| 92 | 6647·63 | 6665·70 | 6683·80 | 6701·93 | 6720·08 | 6738·25 | 6756·45 | 6774·68 | 92 |
| 93 | 6792·92 | 6811·20 | 6829·49 | 6847·82 | 6866·16 | 6884·53 | 6902·93 | 6921·35 | 93 |
| 94 | 6939·79 | 6958·26 | 6976·76 | 6995·28 | 7013·82 | 7032·39 | 7050·98 | 7069·59 | 94 |
| 95 | 7088·24 | 7106·90 | 7125·59 | 7144·31 | 7163·04 | 7181·81 | 7200·60 | 7219·41 | 95 |
| 96 | 7238·25 | 7257·11 | 7275·99 | 7294·91 | 7313·84 | 7332·80 | 7351·79 | 7370·79 | 96 |
| 97 | 7389·83 | 7408·89 | 7427·97 | 7447·08 | 7466·21 | 7485·37 | 7504·55 | 7523·75 | 97 |
| 98 | 7542·98 | 7562·24 | 7581·52 | 7600·82 | 7620·15 | 7639·50 | 7658·88 | 7678·28 | 98 |
| 99 | 7697·71 | 7717·16 | 7736·63 | 7756·13 | 7775·66 | 7795·21 | 7814·78 | 7834·38 | 99 |
| 100 | 7854·00 | 7873·64 | 7893·31 | 7913·00 | 7932·74 | 7952·47 | 7972·25 | 7992·03 | 100 |
| 101 | 8011·86 | 8031·69 | 8051·58 | 8071·46 | 8091·39 | 8111·32 | 8131·30 | 8151·28 | 101 |
| 102 | 8171·30 | 8191·33 | 8211·41 | 8231·49 | 8251·61 | 8271·73 | 8291·91 | 8312·09 | 102 |
| 103 | 8332·31 | 8352·53 | 8372·81 | 8393·08 | 8413·40 | 8433·73 | 8454·09 | 8474·47 | 103 |
| 104 | 8494·89 | 8515·31 | 8535·78 | 8556·25 | 8576·76 | 8597·28 | 8617·85 | 8638·42 | 104 |
| 105 | 8659·03 | ... | 8700·32 | ... | 8741·70 | ... | 8783·18 | ... | 105 |
| 106 | 8824·75 | ... | 8866·43 | ... | 8908·20 | ... | 8950·07 | ... | 106 |
| 107 | 8992·04 | ... | 9034·11 | ... | 9076·28 | ... | 9118·54 | ... | 107 |
| 108 | 9160·91 | ... | 9203·37 | ... | 9245·93 | ... | 9288·58 | ... | 108 |
| 109 | 9331·34 | ... | 9374·19 | ... | 9417·14 | ... | 9460·19 | ... | 109 |
| 110 | 9503·34 | ... | 9546·59 | ... | 9589·93 | ... | 9633·37 | ... | 110 |

Table CXCII.—Areas of Circles—*continued.*

| Diameters. | Areas. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 111 | 9676·91 | ... | 9720·55 | ... | 9764·29 | ... | 9808·12 | ... | 111 |
| 112 | 9852·06 | ... | 9896·09 | ... | 9940·22 | ... | 9984·45 | ... | 112 |
| 113 | 10028·77 | ... | 10073·20 | ... | 10117·72 | ... | 10162·34 | ... | 113 |
| 114 | 10207·06 | ... | 10251·88 | ... | 10296·79 | ... | 10341·80 | ... | 114 |
| 115 | 10386·91 | ... | 10432·12 | ... | 10477·43 | ... | 10522·84 | ... | 115 |
| 116 | 10568·34 | ... | 10613·94 | ... | 10659·65 | ... | 10705·44 | ... | 116 |
| 117 | 10751·34 | ... | 10797·34 | ... | 10843·43 | ... | 10889·62 | ... | 117 |
| 118 | 10935·91 | ... | 10982·30 | ... | 11028·78 | ... | 11075·37 | ... | 118 |
| 119 | 11122·05 | ... | 11168·83 | ... | 11215·71 | ... | 11262·69 | ... | 119 |
| 120 | 11309·76 | ... | 11356·93 | ... | 11404·20 | ... | 11451·57 | ... | 120 |
| 121 | 11499·04 | ... | 11546·61 | ... | 11594·27 | ... | 11642·03 | ... | 121 |
| 122 | 11689·89 | ... | 11737·85 | ... | 11785·91 | ... | 11834·06 | ... | 122 |
| 123 | 11882·32 | ... | 11930·67 | ... | 11979·12 | ... | 12027·66 | ... | 123 |
| 124 | 12076·31 | ... | 12125·05 | ... | 12173·90 | ... | 12222·84 | ... | 124 |
| 125 | 12271·87 | ... | ... | ... | 12370·25 | ... | ... | ... | 125 |
| 126 | 12469·01 | ... | ... | ... | 12568·17 | ... | ... | ... | 126 |
| 127 | 12667·72 | ... | ... | ... | 12767·66 | ... | ... | ... | 127 |
| 128 | 12867·99 | ... | ... | ... | 12968·72 | ... | ... | ... | 128 |
| 129 | 13069·84 | ... | ... | ... | 13171·35 | ... | ... | ... | 129 |
| 130 | 13273·26 | ... | ... | ... | 13375·56 | ... | ... | ... | 130 |
| 131 | 13478·25 | ... | ... | ... | 13581·33 | ... | ... | ... | 131 |
| 132 | 13684·81 | ... | ... | ... | 13788·68 | ... | ... | ... | 132 |

## Table CXCII.—Areas of Circles—*continued.*

| Diameters. | Areas. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 133 | 13892·94 | ... | ... | ... | 13997·60 | ... | ... | ... | 133 |
| 134 | 14102·64 | ... | ... | ... | 14208·08 | ... | ... | ... | 134 |
| 135 | 14313·91 | ... | ... | ... | 14420·14 | ... | ... | ... | 135 |
| 136 | 14526·76 | ... | ... | ... | 14633·77 | ... | ... | ... | 136 |
| 137 | 14741·17 | ... | ... | ... | 14848·97 | ... | ... | ... | 137 |
| 138 | 14957·16 | ... | ... | ... | 15065·74 | ... | ... | ... | 138 |
| 139 | 15174·71 | ... | ... | ... | 15284·08 | ... | ... | ... | 139 |
| 140 | 15393·84 | ... | ... | ... | 15503·99 | ... | ... | ... | 140 |
| 141 | 15614·54 | ... | ... | ... | 15725·48 | ... | ... | ... | 141 |
| 142 | 15836·81 | ... | ... | ... | 15948·53 | ... | ... | ... | 142 |
| 143 | 16060·64 | ... | ... | ... | 16173·15 | ... | ... | ... | 143 |
| 144 | 16286·05 | ... | ... | ... | 16399·35 | ... | ... | ... | 144 |
| 145 | 16513·03 | ... | ... | ... | 16627·11 | ... | ... | ... | 145 |
| 146 | 16741·59 | ... | ... | ... | 16856·45 | ... | ... | ... | 146 |
| 147 | 16971·71 | ... | ... | ... | 17087·36 | ... | ... | ... | 147 |
| 148 | 17203·40 | ... | ... | ... | 17319·84 | ... | ... | ... | 148 |
| 149 | 17436·67 | ... | ... | ... | 17553·89 | ... | ... | ... | 149 |
| 150 | 17671·50 | ... | ... | ... | 17789·51 | ... | ... | ... | 150 |

Where the diameter contains an odd sixteenth, take one fourth of the area of twice the diameter; *e.g.*, area of $2\frac{1}{16} = \frac{\text{Area of } 4\frac{1}{8}}{4}$, &c. Areas of intermediate sizes may also be obtained by shifting the decimal point; *e.g.*, area, 132 = 13684·81; area of 13·2 = 136·8481; area of 1·32 = 1·368481, &c.

## Table CXCIII.—Circumferences of Circles.

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 0 | ·0 | ·3927 | ·7854 | 1·1781 | 1·5708 | 1·9635 | 2·3562 | 2·7489 | 0 |
| 1 | 3·1416 | 3·5343 | 3·9270 | 4·3197 | 4·7124 | 5·1051 | 5·4978 | 5·8905 | 1 |
| 2 | 6·2832 | 6·6759 | 7·0686 | 7·4613 | 7·8540 | 8·2467 | 8·6394 | 9·0321 | 2 |
| 3 | 9·4248 | 9·8175 | 10·2102 | 10·6029 | 10·9956 | 11·3883 | 11·7810 | 12·1737 | 3 |
| 4 | 12·5664 | 12·9591 | 13·3518 | 13·7445 | 14·1372 | 14·5299 | 14·9226 | 15·3153 | 4 |
| 5 | 15·7080 | 16·1007 | 16·4934 | 16·8861 | 17·2788 | 17·6715 | 18·0642 | 18·4569 | 5 |
| 6 | 18·8496 | 19·2423 | 19·6350 | 20·0277 | 20·4204 | 20·8131 | 21·2058 | 21·5985 | 6 |
| 7 | 21·9912 | 22·3839 | 22·7766 | 23·1693 | 23·5620 | 23·9547 | 24·3474 | 24·7401 | 7 |
| 8 | 25·1328 | 25·5255 | 25·9182 | 26·3109 | 26·7036 | 27·0963 | 27·4890 | 27·8817 | 8 |
| 9 | 28·2744 | 28·6671 | 29·0598 | 29·4525 | 29·8452 | 30·2379 | 30·6306 | 31·0233 | 9 |
| 10 | 31·4160 | 31·8087 | 32·2014 | 32·5941 | 32·9868 | 33·3795 | 33·7722 | 34·1649 | 10 |
| 11 | 34·5576 | 34·9503 | 35·3430 | 35·7357 | 36·1284 | 36·5211 | 36·9138 | 37·3065 | 11 |
| 12 | 37·6992 | 38·0919 | 38·4846 | 38·8773 | 39·2700 | 39·6627 | 40·0554 | 40·4481 | 12 |
| 13 | 40·8408 | 41·2335 | 41·6262 | 42·0189 | 42·4116 | 42·8043 | 43·1970 | 43·5897 | 13 |
| 14 | 43·9824 | 44·3751 | 44·7678 | 45·1605 | 45·5532 | 45·9459 | 46·3386 | 46·7313 | 14 |
| 15 | 47·1240 | 47·5167 | 47·9094 | 48·3021 | 48·6948 | 49·0875 | 49·4802 | 49·8729 | 15 |
| 16 | 50·2656 | 50·6583 | 51·0510 | 51·4437 | 51·8364 | 52·2291 | 52·6218 | 53·0145 | 16 |
| 17 | 53·4072 | 53·7999 | 54·1926 | 54·5853 | 54·9780 | 55·3707 | 55·7634 | 56·1561 | 17 |
| 18 | 56·5488 | 56·9415 | 57·3342 | 57·7269 | 58·1196 | 58·5123 | 58·9050 | 59·2977 | 18 |
| 19 | 59·6904 | 60·0831 | 60·4758 | 60·8685 | 61·2612 | 61·6539 | 62·0466 | 62·4393 | 19 |
| 20 | 62·8320 | 63·2247 | 63·6174 | 64·0101 | 64·4028 | 64·7955 | 65·1882 | 65·5809 | 20 |
| 21 | 65·9736 | 66·3663 | 66·7590 | 67·1517 | 67·5444 | 67·9371 | 68·3298 | 68·7225 | 21 |
| 22 | 69·1152 | 69·5079 | 69·9006 | 70·2933 | 70·6860 | 71·0787 | 71·4714 | 71·8641 | 22 |

## Table CXCIII.—Circumferences of Circles—*continued.*

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 23 | 72·2568 | 72·6495 | 73·0422 | 73·4349 | 73·8276 | 74·2203 | 74·6130 | 75·0057 | 23 |
| 24 | 75·3984 | 75·7911 | 76·1838 | 76·5765 | 76·9692 | 77·3619 | 77·7546 | 78·1473 | 24 |
| 25 | 78·5400 | 78·9327 | 79·3254 | 79·7181 | 80·1108 | 80·5035 | 80·8962 | 81·2889 | 25 |
| 26 | 81·6816 | 82·0743 | 82·4670 | 82·8597 | 83·2524 | 83·6451 | 84·0378 | 84·4305 | 26 |
| 27 | 84·8232 | 85·2159 | 85·6086 | 86·0013 | 86·3940 | 86·7867 | 87·1794 | 87·5721 | 27 |
| 28 | 87·9648 | 88·3575 | 88·7502 | 89·1429 | 89·5356 | 89·9283 | 90·3210 | 90·7137 | 28 |
| 29 | 91·1064 | 91·4991 | 91·8918 | 92·2845 | 92·6772 | 93·0699 | 93·4626 | 93·8553 | 29 |
| 30 | 94·2480 | 94·6407 | 95·0334 | 95·4261 | 95·8188 | 96·2115 | 96·6042 | 96·9969 | 30 |
| 31 | 97·3896 | 97·7823 | 98·1750 | 98·5677 | 98·9604 | 99·3531 | 99·7458 | 100·1385 | 31 |
| 32 | 100·5312 | 100·9239 | 101·3166 | 101·7093 | 102·1020 | 102·4947 | 102·8874 | 103·2801 | 32 |
| 33 | 103·6728 | 104·0655 | 104·4582 | 104·8509 | 105·2436 | 105·6363 | 106·0290 | 106·4217 | 33 |
| 34 | 106·814 | 107·207 | 107·600 | 107·992 | 108·385 | 108·778 | 109·171 | 109·563 | 34 |
| 35 | 109·956 | 110·349 | 110·741 | 111·134 | 111·527 | 111·919 | 112·312 | 112·705 | 35 |
| 36 | 113·098 | 113·490 | 113·883 | 114·276 | 114·668 | 115·061 | 115·454 | 115·846 | 36 |
| 37 | 116·239 | 116·632 | 117·025 | 117·417 | 117·810 | 118·203 | 118·595 | 118·988 | 37 |
| 38 | 119·381 | 119·773 | 120·166 | 120·559 | 120·952 | 121·344 | 121·737 | 122·130 | 38 |
| 39 | 122·522 | 122·915 | 123·308 | 123·700 | 124·093 | 124·486 | 124·879 | 125·271 | 39 |
| 40 | 125·664 | 126·057 | 126·449 | 126·842 | 127·235 | 127·627 | 128·020 | 128·413 | 40 |
| 41 | 128·806 | 129·198 | 129·591 | 129·984 | 130·376 | 130·769 | 131·162 | 131·554 | 41 |
| 42 | 131·947 | 132·340 | 132·733 | 133·125 | 133·518 | 133·911 | 134·303 | 134·696 | 42 |
| 43 | 135·089 | 135·481 | 135·874 | 136·267 | 136·660 | 137·052 | 137·445 | 137·838 | 43 |
| 44 | 138·230 | 138·623 | 139·016 | 139·408 | 139·801 | 140·194 | 140·587 | 140·979 | 44 |

## Table CXCIII.—Circumferences of Circles—*continued.*

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 45 | 141·372 | 141·765 | 142·157 | 142·550 | 142·943 | 143·335 | 143·728 | 144·121 | 45 |
| 46 | 144·514 | 144·906 | 145·299 | 145·692 | 146·084 | 146·477 | 146·870 | 147·262 | 46 |
| 47 | 147·655 | 148·048 | 148·441 | 148·833 | 149·226 | 149·619 | 150·011 | 150·404 | 47 |
| 48 | 150·797 | 151·189 | 151·582 | 151·975 | 152·368 | 152·760 | 153·153 | 153·546 | 48 |
| 49 | 153·938 | 154·331 | 154·724 | 155·116 | 155·509 | 155·902 | 156·295 | 156·687 | 49 |
| 50 | 157·080 | 157·473 | 157·865 | 158·258 | 158·651 | 159·043 | 159·436 | 159·829 | 50 |
| 51 | 160·222 | 160·614 | 161·207 | 161·400 | 161·792 | 162·185 | 162·578 | 162·970 | 51 |
| 52 | 163·363 | 163·756 | 164·149 | 164·541 | 164·934 | 165·327 | 165·719 | 166·112 | 52 |
| 53 | 166·505 | 166·897 | 167·290 | 167·683 | 168·076 | 168·468 | 168·861 | 169·254 | 53 |
| 54 | 169·646 | 170·039 | 170·432 | 170·824 | 171·217 | 171·610 | 172·003 | 172·395 | 54 |
| 55 | 172·788 | 173·181 | 173·573 | 173·966 | 174·359 | 174·751 | 175·144 | 175·537 | 55 |
| 56 | 175·930 | 176·322 | 176·715 | 177·108 | 177·500 | 177·893 | 178·286 | 178·678 | 56 |
| 57 | 179·071 | 179·464 | 179·857 | 180·249 | 180·642 | 181·035 | 181·427 | 181·820 | 57 |
| 58 | 182·213 | 182·605 | 182·998 | 183·391 | 183·784 | 184·176 | 184·569 | 184·962 | 58 |
| 59 | 185·354 | 185·747 | 186·140 | 186·532 | 186·925 | 187·318 | 187·711 | 188·103 | 59 |
| 60 | 188·496 | 188·889 | 189·281 | 189·674 | 190·067 | 190·459 | 190·852 | 191·245 | 60 |
| 61 | 191·638 | 192·030 | 192·423 | 192·816 | 193·208 | 193·601 | 193·994 | 194·396 | 61 |
| 62 | 194·779 | 195·172 | 195·565 | 195·957 | 196·350 | 196·743 | 197·135 | 197·528 | 62 |
| 63 | 197·921 | 198·313 | 198·706 | 199·099 | 199·492 | 199·884 | 200·277 | 200·670 | 63 |
| 64 | 201·062 | 201·455 | 201·848 | 202·240 | 202·633 | 203·026 | 203·419 | 203·811 | 64 |
| 65 | 204·204 | 204·597 | 204·989 | 205·382 | 205·775 | 206·167 | 206·560 | 206·953 | 65 |
| 66 | 207·346 | 207·738 | 208·131 | 208·524 | 208·916 | 209·309 | 209·702 | 210·094 | 66 |

## Table CXCIII.—Circumferences of Circles—*continued.*

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 67 | 210·487 | 210·880 | 211·273 | 211·665 | 212·058 | 212·451 | 212·843 | 213·236 | 67 |
| 68 | 213·629 | 214·021 | 214·414 | 214·807 | 215·200 | 215·592 | 215·985 | 216·378 | 68 |
| 69 | 216·770 | 217·163 | 217·556 | 217·948 | 218·341 | 218·734 | 219·127 | 219·519 | 69 |
| 70 | 219·912 | 220·305 | 220·697 | 221·090 | 221·483 | 221·875 | 222·268 | 222·661 | 70 |
| 71 | 223·054 | 223·446 | 223·839 | 224·232 | 224·624 | 225·017 | 225·410 | 225·802 | 71 |
| 72 | 226·195 | 226·588 | 226·981 | 227·373 | 227·766 | 228·158 | 228·551 | 228·944 | 72 |
| 73 | 229·337 | 229·729 | 230·122 | 230·515 | 230·908 | 231·300 | 231·693 | 232·086 | 73 |
| 74 | 232·478 | 232·871 | 233·264 | 233·656 | 234·049 | 234·442 | 234·835 | 235·227 | 74 |
| 75 | 235·620 | 236·013 | 236·405 | 236·798 | 237·191 | 237·583 | 237·976 | 238·369 | 75 |
| 76 | 238·762 | 239·154 | 239·547 | 239·940 | 240·332 | 240·725 | 241·118 | 241·510 | 76 |
| 77 | 241·903 | 242·296 | 242·689 | 243·081 | 243·474 | 243·867 | 244·259 | 244·652 | 77 |
| 78 | 245·045 | 245·437 | 245·830 | 246·223 | 246·616 | 247·008 | 247·401 | 247·794 | 78 |
| 79 | 248·186 | 248·579 | 248·972 | 249·364 | 249·757 | 250·150 | 250·543 | 250·935 | 79 |
| 80 | 251·328 | 251·721 | 252·113 | 252·506 | 252·899 | 253·291 | 253·684 | 254·077 | 80 |
| 81 | 254·470 | 254·862 | 255·255 | 255·648 | 256·040 | 256·433 | 256·826 | 257·218 | 81 |
| 82 | 257·611 | 258·004 | 258·397 | 258·789 | 259·182 | 259·575 | 259·967 | 260·360 | 82 |
| 83 | 260·753 | 261·145 | 261·538 | 261·931 | 262·324 | 262·716 | 263·109 | 263·502 | 83 |
| 84 | 263·894 | 264·287 | 264·680 | 265·072 | 265·465 | 265·858 | 266·251 | 266·643 | 84 |
| 85 | 267·036 | 267·429 | 267·821 | 268·214 | 268·607 | 268·999 | 269·392 | 269·785 | 85 |
| 86 | 270·178 | 270·570 | 270·963 | 271·356 | 271·748 | 272·141 | 272·534 | 272·926 | 86 |
| 87 | 273·319 | 273·712 | 274·105 | 274·497 | 274·890 | 275·283 | 275·675 | 276·068 | 87 |
| 88 | 276·461 | 276·853 | 277·246 | 277·629 | 278·032 | 278·424 | 278·817 | 279·210 | 88 |

Table CXCIII.—Circumferences of Circles—*continued.*

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 89 | 279·602 | 279·995 | 280·388 | 280·780 | 281·173 | 281·566 | 281·959 | 282·351 | 89 |
| 90 | 282·744 | 283·137 | 283·529 | 283·922 | 284·315 | 284·707 | 285·100 | 285·493 | 90 |
| 91 | 285·886 | 286·278 | 286·671 | 287·064 | 287·456 | 287·849 | 288·242 | 288·634 | 91 |
| 92 | 289·027 | 289·420 | 289·813 | 290·205 | 290·598 | 290·991 | 291·383 | 291·776 | 92 |
| 93 | 292·169 | 292·562 | 292·954 | 293·347 | 293·740 | 294·132 | 294·525 | 294·918 | 93 |
| 94 | 295·310 | 295·703 | 296·096 | 296·488 | 296·881 | 297·274 | 297·667 | 298·059 | 94 |
| 95 | 298·452 | 298·845 | 299·237 | 299·630 | 300·023 | 300·415 | 300·808 | 301·201 | 95 |
| 96 | 301·594 | 301·986 | 302·379 | 302·772 | 303·164 | 303·557 | 303·950 | 304·342 | 96 |
| 97 | 304·735 | 305·128 | 305·521 | 305·913 | 306·306 | 306·699 | 307·091 | 307·484 | 97 |
| 98 | 307·877 | 308·270 | 308·662 | 309·055 | 309·448 | 309·840 | 310·233 | 310·626 | 98 |
| 99 | 311·018 | 311·411 | 311·804 | 312·196 | 312·589 | 312·982 | 313·375 | 313·767 | 99 |
| 100 | 314·160 | 314·553 | 314·945 | 315·338 | 315·731 | 316·123 | 316·516 | 316·909 | 100 |
| 101 | 317·302 | 317·694 | 318·087 | 318·480 | 318·872 | 319·265 | 319·658 | 320·050 | 101 |
| 102 | 320·443 | 320·836 | 321·229 | 321·621 | 322·014 | 322·407 | 322·799 | 323·192 | 102 |
| 103 | 323·585 | 323·977 | 324·370 | 324·763 | 325·156 | 325·548 | 325·941 | 326·334 | 103 |
| 104 | 326·726 | 327·119 | 327·512 | 327·904 | 328·297 | 328·690 | 329·083 | 329·475 | 104 |
| 105 | 329·868 | ... | 330·653 | ... | 331·439 | ... | 332·224 | ... | 105 |
| 106 | 333·010 | ... | 333·795 | ... | 334·580 | ... | 335·366 | ... | 106 |
| 107 | 336·151 | ... | 336·937 | ... | 337·722 | ... | 338·507 | ... | 107 |
| 108 | 339·293 | ... | 340·078 | ... | 340·864 | ... | 341·649 | ... | 108 |
| 109 | 342·434 | ... | 343·220 | ... | 344·005 | ... | 344·791 | ... | 109 |
| 110 | 345·576 | ... | 346·361 | ... | 347·147 | ... | 347·932 | ... | 110 |

## Table CXCIII.—Circumferences of Circles—*continued.*

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 111 | 348·718 | ... | 349·503 | ... | 350·288 | ... | 351·074 | ... | 111 |
| 112 | 351·859 | ... | 352·645 | ... | 353·430 | ... | 354·215 | ... | 112 |
| 113 | 355·001 | ... | 355·786 | ... | 356·572 | ... | 357·357 | ... | 113 |
| 114 | 358·142 | ... | 358·928 | ... | 359·713 | ... | 360·499 | ... | 114 |
| 115 | 361·284 | ... | 362·069 | ... | 362·855 | ... | 363·640 | ... | 115 |
| 116 | 364·426 | ... | 365·211 | ... | 365·996 | ... | 366·782 | ... | 116 |
| 117 | 367·567 | ... | 368·353 | ... | 369·138 | ... | 369·923 | ... | 117 |
| 118 | 370·709 | ... | 371·494 | ... | 372·280 | ... | 373·065 | ... | 118 |
| 119 | 373·850 | ... | 374·636 | ... | 375·421 | ... | 376·207 | ... | 119 |
| 120 | 376·992 | ... | 377·777 | ... | 378·563 | ... | 379·348 | ... | 120 |
| 121 | 380·134 | ... | 380·919 | ... | 381·704 | ... | 382·490 | ... | 121 |
| 122 | 383·275 | ... | 384·061 | ... | 384·846 | ... | 385·631 | ... | 122 |
| 123 | 386·417 | ... | 387·202 | ... | 387·988 | ... | 388·773 | ... | 123 |
| 124 | 389·558 | ... | 390·344 | ... | 391·129 | ... | 391·915 | ... | 124 |
| 125 | 392·700 | ... | ... | ... | 394·271 | ... | ... | ... | 125 |
| 126 | 395·842 | ... | ... | ... | 397·412 | ... | ... | ... | 126 |
| 127 | 398·983 | ... | ... | ... | 400·554 | ... | ... | ... | 127 |
| 128 | 402·125 | ... | ... | ... | 403·696 | ... | ... | ... | 128 |
| 129 | 405·266 | ... | ... | ... | 406·837 | ... | ... | ... | 129 |
| 130 | 408·408 | ... | ... | ... | 409·979 | ... | ... | ... | 130 |
| 131 | 411·550 | ... | ... | ... | 413·120 | ... | ... | ... | 131 |
| 132 | 414·691 | ... | ... | ... | 416·262 | ... | ... | ... | 132 |

## Table CXCIII.—Circumferences of Circles—*continued.*

| Diameters. | Circumferences. | | | | | | | | Diameters. |
|---|---|---|---|---|---|---|---|---|---|
| | ·0 | ·⅛ | ·¼ | ·⅜ | ·½ | ·⅝ | ·¾ | ·⅞ | |
| 133 | 417·833 | ... | ... | ... | 419·404 | ... | ... | ... | 133 |
| 134 | 420·974 | ... | ... | ... | 422·545 | ... | ... | ... | 134 |
| 135 | 424·116 | ... | ... | ... | 425·687 | ... | ... | ... | 135 |
| 136 | 427·258 | ... | ... | ... | 428·828 | ... | ... | ... | 136 |
| 137 | 430·399 | ... | ... | ... | 431·970 | ... | ... | ... | 137 |
| 138 | 433·541 | ... | ... | ... | 435·112 | ... | ... | ... | 138 |
| 139 | 436·682 | ... | ... | ... | 438·253 | ... | ... | ... | 139 |
| 140 | 439·824 | ... | ... | ... | 441·395 | ... | ... | ... | 140 |
| 141 | 442·966 | ... | ... | ... | 444·536 | ... | ... | ... | 141 |
| 142 | 446·107 | ... | ... | ... | 447·678 | ... | ... | ... | 142 |
| 143 | 449·249 | ... | ... | ... | 450·820 | ... | ... | ... | 143 |
| 144 | 452·390 | ... | ... | ... | 453·961 | ... | ... | ... | 144 |
| 145 | 455·532 | ... | ... | ... | 457·103 | ... | ... | ... | 145 |
| 146 | 458·674 | ... | ... | ... | 460·244 | ... | ... | ... | 146 |
| 147 | 461·815 | ... | ... | ... | 463·386 | ... | ... | ... | 147 |
| 148 | 464·957 | ... | ... | ... | 466·528 | ... | ... | ... | 148 |
| 149 | 468·098 | ... | ... | ... | 469·669 | ... | ... | ... | 149 |
| 150 | 471·240 | ... | ... | ... | 472·811 | ... | ... | ... | 150 |

## Table CXCIV.—Areas and Circumferences of Small Circles.

| Diameter. | Area. | Circum. | Diameter. | Area. | Circum. |
|---|---|---|---|---|---|
| 1/32 | ·000767 | ·09817 | 1 1/32 | ·83525 | 3·2398 |
| 1/16 | ·003068 | ·19635 | 1 1/16 | ·88665 | 3·3379 |
| 3/32 | ·006903 | ·29452 | 1 3/32 | ·93956 | 3·4361 |
| 1/8 | ·012272 | ·39270 | 1 1/8 | ·99402 | 3·5343 |
| 5/32 | ·019175 | ·49087 | 1 5/32 | 1·0500 | 3·6325 |
| 3/16 | ·027612 | ·58905 | 1 3/16 | 1·1075 | 3·7306 |
| 7/32 | ·037583 | ·68722 | 1 7/32 | 1·1666 | 3·8288 |
| 1/4 | ·049087 | ·78540 | 1 1/4 | 1·2272 | 3·9270 |
| 9/32 | ·062126 | ·88357 | 1 9/32 | 1·2893 | 4·0252 |
| 5/16 | ·076699 | ·98175 | 1 5/16 | 1·3530 | 4·1233 |
| 11/32 | ·092806 | 1·0799 | 1 11/32 | 1·4182 | 4·2215 |
| 3/8 | ·11045 | 1·1781 | 1 3/8 | 1·4849 | 4·3197 |
| 13/32 | ·12962 | 1·2763 | 1 13/32 | 1·5531 | 4·4179 |
| 7/16 | ·15033 | 1·3744 | 1 7/16 | 1·6229 | 4·5160 |
| 15/32 | ·17257 | 1·4726 | 1 15/32 | 1·6943 | 4·6142 |
| 1/2 | ·19635 | 1·5708 | 1 1/2 | 1·7671 | 4·7124 |
| 17/32 | ·22166 | 1·6690 | 1 17/32 | 1·8415 | 4·8106 |
| 9/16 | ·24850 | 1·7671 | 1 9/16 | 1·9175 | 4·9087 |
| 19/32 | ·27688 | 1·8653 | 1 19/32 | 1·9949 | 5·0069 |
| 5/8 | ·30680 | 1·9635 | 1 5/8 | 2·0739 | 5·1051 |
| 21/32 | ·33824 | 2·0617 | 1 21/32 | 2·1545 | 5·2033 |
| 11/16 | ·37122 | 2·1598 | 1 11/16 | 2·2365 | 5·3014 |
| 23/32 | ·40574 | 2·2580 | 1 23/32 | 2·3201 | 5·3996 |
| 3/4 | ·44179 | 2·3562 | 1 3/4 | 2·4053 | 5·4978 |
| 25/32 | ·47937 | 2·4544 | 1 25/32 | 2·4919 | 5·5959 |
| 13/16 | ·51849 | 2·5525 | 1 13/16 | 2·5801 | 5·6941 |
| 27/32 | ·55914 | 2·6507 | 1 27/32 | 2·6699 | 5·7923 |
| 7/8 | ·60132 | 2·7489 | 1 7/8 | 2·7612 | 5·8905 |
| 29/32 | ·64504 | 2·8471 | 1 29/32 | 2·8540 | 5·9886 |
| 15/16 | ·69029 | 2·9452 | 1 15/16 | 2·9483 | 6·0868 |
| 31/32 | ·73708 | 3·0434 | 1 31/32 | 3·0442 | 6·1850 |
| 1 | ·78540 | 3·1416 | 2 | 3·1416 | 6·2832 |

## Table CXCV.—Spheres, Volume and Surface of.

(Volume $=\frac{\pi d^3}{6}=\frac{d^3}{1\cdot91}$ ; surface $=\pi d^2$.)

| Dia. | Surface. | Volume. | Dia. | Surface. | Volume. | Dia. | Surface. | Volume. |
|---|---|---|---|---|---|---|---|---|
| 1 | 3·142 | 0·524 | 41 | 5281·0 | 36,084 | 81 | 20,612 | 278,241 |
| 2 | 12·546 | 4·189 | 42 | 5541·8 | 38,785 | 82 | 21,120 | 288,640 |
| 3 | 28·274 | 14·136 | 43 | 5808·8 | 41,627 | 83 | 21,633 | 299,361 |
| 4 | 50·266 | 33·508 | 44 | 6082·1 | 44,597 | 84 | 22,146 | 310,316 |
| 5 | 78·540 | 65·445 | 45 | 6361·7 | 47,707 | 85 | 22,704 | 321,531 |
| 6 | 113·098 | 113·09 | 46 | 6647·6 | 50,961 | 86 | 23,227 | 333,013 |
| 7 | 153·938 | 179·58 | 47 | 6939·8 | 54,356 | 87 | 23,753 | 344,764 |
| 8 | 201·062 | 268·06 | 48 | 7238·3 | 57,900 | 88 | 24,328 | 356,791 |
| 9 | 254·47 | 381·67 | 49 | 7543·0 | 61,596 | 89 | 24,860 | 369,089 |
| 10 | 314·16 | 523·56 | 50 | 7854·0 | 65,445 | 90 | 25,434 | 381,675 |
| 11 | 380·13 | 696·86 | 51 | 8171·3 | 69,450 | 91 | 26,016 | 394,540 |
| 12 | 452·39 | 904·71 | 52 | 8494·9 | 73,612 | 92 | 26,601 | 407,690 |
| 13 | 530·93 | 1150·3 | 53 | 8824·8 | 77,946 | 93 | 27,150 | 421,130 |
| 14 | 615·75 | 1436·7 | 54 | 9160·9 | 82,442 | 94 | 27,742 | 434,860 |
| 15 | 706·86 | 1767·0 | 55 | 9503·3 | 87,105 | 95 | 28,338 | 448,887 |
| 16 | 804·25 | 2144·5 | 56 | 9852·1 | 91,945 | 96 | 28,939 | 463,212 |
| 17 | 907·92 | 2572·2 | 57 | 10,207 | 96,960 | 97 | 29,543 | 477,840 |
| 18 | 1017·9 | 3053·4 | 58 | 10,568 | 102,152 | 98 | 30,150 | 492,771 |
| 19 | 1134·1 | 3591·1 | 59 | 10,936 | 107,523 | 99 | 30,787 | 508,010 |
| 20 | 1256·6 | 4188·5 | 60 | 11,310 | 113,089 | 100 | 31,416 | 523,560 |
| 21 | 1385·5 | 4848 | 61 | 11,690 | 118,847 | 101 | 32,036 | 539,424 |
| 22 | 1520·5 | 5575 | 62 | 12,076 | 124,780 | 102 | 32,685 | 555,602 |
| 23 | 1661·9 | 6370 | 63 | 12,469 | 130,914 | 103 | 33,329 | 572,109 |
| 24 | 1809·6 | 7237 | 64 | 12,868 | 137,248 | 104 | 33,980 | 588,932 |
| 25 | 1963·5 | 8181 | 65 | 13,273 | 143,783 | 105 | 34,636 | 606,086 |
| 26 | 2123·7 | 9202 | 66 | 13,685 | 150,521 | 106 | 35,299 | 623,570 |
| 27 | 2290·2 | 10,305 | 67 | 14,103 | 157,466 | 107 | 35,968 | 641,384 |
| 28 | 2463·0 | 11,490 | 68 | 14,527 | 164,623 | 108 | 36,644 | 659,535 |
| 29 | 2642·1 | 12,769 | 69 | 14,957 | 172,000 | 109 | 37,325 | 678,025 |
| 30 | 2827·4 | 14,136 | 70 | 15,394 | 179,581 | 110 | 38,013 | 696,860 |
| 31 | 3019·1 | 15,597 | 71 | 15,837 | 187,387 | 111 | 38,708 | 716,037 |
| 32 | 3217·0 | 17,152 | 72 | 16,286 | 195,418 | 112 | 39,408 | 735,560 |
| 33 | 3421·2 | 18,812 | 73 | 16,742 | 203,674 | 113 | 40,116 | 755,440 |
| 34 | 3631·7 | 20,523 | 74 | 17,204 | 212,193 | 114 | 40,828 | 775,676 |
| 35 | 3848·5 | 22,377 | 75 | 17,672 | 220,878 | 115 | 41,548 | 796,270 |
| 36 | 4071·5 | 24,372 | 76 | 18,146 | 229,827 | 116 | 42,273 | 817,223 |
| 37 | 4300·9 | 26,466 | 77 | 18,826 | 239,023 | 117 | 43,005 | 838,535 |
| 38 | 4536·5 | 28,675 | 78 | 19,114 | 248,456 | 118 | 43,744 | 860,227 |
| 39 | 4778·4 | 31,057 | 79 | 19,607 | 258,135 | 119 | 44,488 | 882,282 |
| 40 | 5026·6 | 33,508 | 80 | 20,106 | 268,063 | 120 | 45,240 | 904,712 |

## Table CXCV.—Spheres, Volume and Surface of—*continued.*

(Volume $=\frac{\pi d^3}{6}=\frac{d^3}{1·91}$; surface $=\pi d^2$.)

| Dia. | Surface. | Volume. | Dia. | Surface. | Volume. | Dia. | Surface. | Volume. |
|---|---|---|---|---|---|---|---|---|
| 121 | 45,996 | 927,519 | 131 | 53,913 | 1,177,000 | 141 | 62,460 | 1,467,644 |
| 122 | 46,760 | 940,413 | 132 | 54,740 | 1,204,172 | 142 | 63,348 | 1,500,000 |
| 123 | 47,529 | 974,276 | 133 | 55,572 | 1,231,750 | 143 | 64,244 | 1,531,100 |
| 124 | 48,305 | 998,230 | 134 | 56,412 | 1,260,000 | 144 | 65,144 | 1,563,342 |
| 125 | 49,088 | 1,022,577 | 135 | 57,256 | 1,288,152 | 145 | 66,052 | 1,596,131 |
| 126 | 49,876 | 1,047,320 | 136 | 58,108 | 1,317,000 | 146 | 66,968 | 1,629,390 |
| 127 | 50,672 | 1,072,452 | 137 | 58,965 | 1,346,251 | 147 | 67,888 | 1,663,100 |
| 128 | 51,472 | 1,098,000 | 138 | 59,829 | 1,376,000 | 148 | 68,813 | 1,700,000 |
| 129 | 52,280 | 1,123,916 | 139 | 60,700 | 1,406,083 | 149 | 69,748 | 1,731,900 |
| 130 | 53,092 | 1,150,262 | 140 | 61,576 | 1,436,650 | 150 | 70,688 | 1,767,000 |

*Note.—Cones* have half the volume and 0·809 of the surface of corresponding spheres, that is, cones having a diameter of base and height each $=d$, volume $=d^3 \div 3·82$, surface $=d^2 \times 2·5416$.

## STOWAGE.

### Miscellaneous Weights and Measures.

### Coal.

1 sack = 224 lbs.
1 chald. = 12 sacks.
1 Newcastle chald. = 23½ sacks.
1 keel = 8 Newcastle chald.

| | | | | | |
|---|---|---|---|---|---|
| Welsh . | 42·7 | cubic ft. to ton. | Lancashire | 45·2 | cubic ft. to ton. |
| Newcastle . | 45·3 | ,, ,, | Scotch . | 42 | ,, ,, |
| S. Yorkshire | 46 | ,, ,, | American | 42·4 | ,, ,, |

### Grain.

480 lbs. = 1 quarter (approximate).
Oats 56 bushels = 1 ton ,,
,, 1 ,, = 40 lbs. ,,
Wheat 37⅓ ,, = 1 ton ,,
,, 1 ,, = 60 lbs. ,,
Maize 40 ,, = 1 ton ,,

### Cotton (American).

4½ bales = 1 ton (measurement, approximate).
2⅛ to 2¼ bales = 1 ton oil-cake or meal ,,
1 bale = 20 bushels wheat ,,

### Table CXCVI.—Squares, Cubes, Square Roots, Cube Roots, and Reciprocals of all Integer Numbers from 1 to 2200.

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 1 | 1 | 1 | 1·0000000 | 1·0000000 | 1·000000000 |
| 2 | 4 | 8 | 1·4142136 | 1·2599210 | ·500000000 |
| 3 | 9 | 27 | 1·7320508 | 1·4422496 | ·333333333 |
| 4 | 16 | 64 | 2·0000000 | 1·5874011 | ·250000000 |
| 5 | 25 | 125 | 2·2360680 | 1·7099759 | ·200000000 |
| 6 | 36 | 216 | 2·4494897 | 1·8171206 | ·166666667 |
| 7 | 49 | 343 | 2·6457513 | 1·9129312 | ·142857143 |
| 8 | 64 | 512 | 2·8284271 | 2·0000000 | ·125000000 |
| 9 | 81 | 729 | 3·0000000 | 2·0800837 | ·111111111 |
| 10 | 100 | 1000 | 3·1622777 | 2·1544347 | ·100000000 |
| 11 | 121 | 1331 | 3·3166248 | 2·2239801 | ·090909091 |
| 12 | 144 | 1728 | 3·4641016 | 2·2894286 | ·083333333 |
| 13 | 169 | 2197 | 3·6055513 | 2·3513347 | ·076923077 |
| 14 | 196 | 2744 | 3·7416574 | 2·4101422 | ·071428571 |
| 15 | 225 | 3375 | 3·8729833 | 2·4662121 | ·066666667 |
| 16 | 256 | 4096 | 4·0000000 | 2·5198421 | ·062500000 |
| 17 | 289 | 4913 | 4·1231056 | 2·5712816 | ·058823529 |
| 18 | 324 | 5832 | 4·2426407 | 2·6207414 | ·055555556 |
| 19 | 361 | 6859 | 4·3588989 | 2·6684016 | ·052631579 |
| 20 | 400 | 8000 | 4·4721360 | 2·7144177 | ·050000000 |
| 21 | 441 | 9261 | 4·5825757 | 2·7589243 | ·047619048 |
| 22 | 484 | 10648 | 4·6904158 | 2·8020393 | ·045454545 |
| 23 | 529 | 12167 | 4·7958315 | 2·8438670 | ·043478261 |
| 24 | 576 | 13824 | 4·8989795 | 2·8844991 | ·041666667 |
| 25 | 625 | 15625 | 5·0000000 | 2·9240177 | ·040000000 |
| 26 | 676 | 17576 | 5·0990195 | 2·9624960 | ·038461538 |
| 27 | 729 | 19683 | 5·1961524 | 3·0000000 | ·037037037 |
| 28 | 784 | 21952 | 5·2915026 | 3·0365889 | ·035714286 |
| 29 | 841 | 24389 | 5·3851648 | 3·0723168 | ·034482759 |
| 30 | 900 | 27000 | 5·4772256 | 3·1072325 | ·033333333 |
| 31 | 961 | 29791 | 5·5677644 | 3·1413806 | ·032258065 |
| 32 | 1024 | 32768 | 5·6568542 | 3·1748021 | ·031250000 |
| 33 | 1089 | 35937 | 5·7445626 | 3·2075343 | ·030303030 |
| 34 | 1156 | 39304 | 5·8309519 | 3·2396118 | ·029411765 |
| 35 | 1225 | 42875 | 5·9160798 | 3·2710663 | ·028571429 |
| 36 | 1296 | 46656 | 6·0000000 | 3·3019272 | ·027777778 |
| 37 | 1369 | 50653 | 6·0827625 | 3·3322218 | ·027027027 |
| 38 | 1444 | 54872 | 6·1644140 | 3·3619754 | ·026315789 |
| 39 | 1521 | 59319 | 6·2449980 | 3·3912114 | ·025641026 |
| 40 | 1600 | 64000 | 6·3245553 | 3·4199519 | ·025000000 |
| 41 | 1681 | 68921 | 6·4031242 | 3·4482172 | ·024390244 |
| 42 | 1764 | 74088 | 6·4807407 | 3·4760266 | ·023809524 |
| 43 | 1849 | 79507 | 6·5574385 | 3·5033981 | ·023255814 |
| 44 | 1936 | 85184 | 6·6332496 | 3·5303483 | ·022727273 |
| 45 | 2025 | 91125 | 6·7082039 | 3·5568933 | ·022222222 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 46 | 2116 | 97336 | 6·7823300 | 3·5830479 | ·021739130 |
| 47 | 2209 | 103823 | 6·8556546 | 3·6088261 | ·021276600 |
| 48 | 2304 | 110592 | 6·9282032 | 3·6342411 | ·020833333 |
| 49 | 2401 | 117649 | 7·0000000 | 3·6593057 | ·020408163 |
| 50 | 2500 | 125000 | 7·0710678 | 3·6840314 | ·020000000 |
| 51 | 2601 | 132651 | 7·1414284 | 3·7084298 | ·019607843 |
| 52 | 2704 | 140608 | 7·2111026 | 3·7325111 | ·019230769 |
| 53 | 2809 | 148877 | 7·2801099 | 3·7562858 | ·018867925 |
| 54 | 2916 | 157464 | 7·3484692 | 3·7797631 | ·018518519 |
| 55 | 3025 | 166375 | 7·4161985 | 3·8029525 | ·018181818 |
| 56 | 3136 | 175616 | 7·4833148 | 3·8258624 | ·017857143 |
| 57 | 3249 | 185193 | 7·5493344 | 3·8485011 | ·017543860 |
| 58 | 3364 | 195112 | 7·6157731 | 3·8708766 | ·017241379 |
| 59 | 3481 | 205379 | 7·6811457 | 3·8929965 | ·016949153 |
| 60 | 3600 | 216000 | 7·7459667 | 3·9148676 | ·016666667 |
| 61 | 3721 | 226981 | 7·8102497 | 3·9364972 | ·016393443 |
| 62 | 3844 | 238328 | 7·8740079 | 3·9578915 | ·016129032 |
| 63 | 3969 | 250047 | 7·9372539 | 3·9790571 | ·015873016 |
| 64 | 4096 | 262144 | 8·0000000 | 4·0000000 | ·015625000 |
| 65 | 4225 | 274625 | 8·0622577 | 4·0207256 | ·015384615 |
| 66 | 4356 | 287496 | 8·1240384 | 4·0412401 | ·015151515 |
| 67 | 4489 | 300763 | 8·1853528 | 4·0615480 | ·014925373 |
| 68 | 4624 | 314432 | 8·2462113 | 4·0816551 | ·014705882 |
| 69 | 4761 | 328509 | 8·3066239 | 4·1015661 | ·014492754 |
| 70 | 4900 | 343000 | 8·3666003 | 4·1212853 | ·014285714 |
| 71 | 5041 | 357911 | 8·4261498 | 4·1408178 | ·014084507 |
| 72 | 5184 | 373248 | 8·4852814 | 4·1601676 | ·013888889 |
| 73 | 5329 | 389017 | 8·5440037 | 4·1793392 | ·013698630 |
| 74 | 5476 | 405224 | 8·6023253 | 4·1983364 | ·013513514 |
| 75 | 5625 | 421875 | 8·6602540 | 4·2171633 | ·013333333 |
| 76 | 5776 | 438976 | 8·7177979 | 4·2358236 | ·013157895 |
| 77 | 5929 | 456533 | 8·7749644 | 4·2543210 | ·012987013 |
| 78 | 6084 | 474552 | 8·8317609 | 4·2726586 | ·012820513 |
| 79 | 6241 | 493039 | 8·8881944 | 4·2908404 | ·012658228 |
| 80 | 6400 | 512000 | 8·9442719 | 4·3088695 | ·012500000 |
| 81 | 6561 | 531441 | 9·0000000 | 4·3267487 | ·012345679 |
| 82 | 6724 | 551368 | 9·0553851 | 4·3444815 | ·012195122 |
| 83 | 6889 | 571787 | 9·1104336 | 4·3620707 | ·012048193 |
| 84 | 7056 | 592704 | 9·1651514 | 4·3795191 | ·011904762 |
| 85 | 7225 | 614125 | 9·2195445 | 4·3968296 | ·011764706 |
| 86 | 7396 | 636056 | 9·2736185 | 4·4140049 | ·011627907 |
| 87 | 7569 | 658503 | 9·3273791 | 4·4310476 | ·011494253 |
| 88 | 7744 | 681472 | 9·3808315 | 4·4479602 | ·011363636 |
| 89 | 7921 | 704969 | 9·4339811 | 4·4647451 | ·011235955 |
| 90 | 8100 | 729000 | 9·4868330 | 4·4814047 | ·011111111 |
| 91 | 8281 | 753571 | 9·5393920 | 4·4979414 | ·010989011 |
| 92 | 8464 | 778688 | 9·5916630 | 4·5143574 | ·010869565 |
| 93 | 8649 | 804357 | 9·6436508 | 4·5306549 | ·010752688 |
| 94 | 8836 | 830584 | 9·6953597 | 4·5468359 | ·010638298 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 95 | 9025 | 857375 | 9·7467943 | 4·5629026 | ·010526316 |
| 96 | 9216 | 884736 | 9·7979590 | 4·5788570 | ·010416667 |
| 97 | 9409 | 912673 | 9·8488578 | 4·5947009 | ·010309278 |
| 98 | 9604 | 941192 | 9·8994949 | 4·6104363 | ·010204082 |
| 99 | 9801 | 970299 | 9·9498744 | 4·6260650 | ·010101010 |
| 100 | 10000 | 1000000 | 10·0000000 | 4·6415888 | ·010000000 |
| 101 | 10201 | 1030301 | 10·0498756 | 4·6570095 | ·009900990 |
| 102 | 10404 | 1061208 | 10·0995049 | 4·6723287 | ·009803922 |
| 103 | 10609 | 1092727 | 10·1488916 | 4·6875482 | ·009708738 |
| 104 | 10816 | 1124864 | 10·1980390 | 4·7026694 | ·009615385 |
| 105 | 11025 | 1157625 | 10·2469508 | 4·7176940 | ·009523810 |
| 106 | 11236 | 1191016 | 10·2956301 | 4·7326235 | ·009433962 |
| 107 | 11449 | 1225043 | 10·3440804 | 4·7474594 | ·009345794 |
| 108 | 11664 | 1259712 | 10·3923048 | 4·7622032 | ·009259259 |
| 109 | 11881 | 1295029 | 10·4403065 | 4·7768562 | ·009174312 |
| 110 | 12100 | 1331000 | 10·4880885 | 4·7914199 | ·009090909 |
| 111 | 12321 | 1367631 | 10·5356538 | 4·8058955 | ·009009009 |
| 112 | 12544 | 1404928 | 10·5830052 | 4·8202845 | ·008928571 |
| 113 | 12769 | 1442897 | 10·6301458 | 4·8345881 | ·008849558 |
| 114 | 12996 | 1481544 | 10·6770783 | 4·8488076 | ·008771930 |
| 115 | 13225 | 1520875 | 10·7238053 | 4·8629442 | ·008695652 |
| 116 | 13456 | 1560896 | 10·7703296 | 4·8769990 | ·008620690 |
| 117 | 13689 | 1601613 | 10·8166538 | 4·8909732 | ·008547009 |
| 118 | 13924 | 1643032 | 10·8627805 | 4·9048681 | ·008474576 |
| 119 | 14161 | 1685159 | 10·9087121 | 4·9186847 | ·008403361 |
| 120 | 14400 | 1728000 | 10·9544512 | 4·9324242 | ·008333333 |
| 121 | 14641 | 1771561 | 11·0000000 | 4·9460874 | ·008264463 |
| 122 | 14884 | 1815848 | 11·0453610 | 4·9596757 | ·008196721 |
| 123 | 15129 | 1860867 | 11·0905365 | 4·9731898 | ·008130081 |
| 124 | 15376 | 1906624 | 11·1355287 | 4·9866310 | ·008064516 |
| 125 | 15625 | 1953125 | 11·1803399 | 5·0000000 | ·008000000 |
| 126 | 15876 | 2000376 | 11·2249722 | 5·0132979 | ·007936508 |
| 127 | 16129 | 2048383 | 11·2694277 | 5·0265257 | ·007874016 |
| 128 | 16384 | 2097152 | 11·3137085 | 5·0396842 | ·007812500 |
| 129 | 16641 | 2146689 | 11·3578167 | 5·0527743 | ·007751938 |
| 130 | 16900 | 2197000 | 11·4017543 | 5·0657970 | ·007692308 |
| 131 | 17161 | 2248091 | 11·4455231 | 5·0787531 | ·007633588 |
| 132 | 17424 | 2299968 | 11·4891253 | 5·0916434 | ·007575758 |
| 133 | 17689 | 2352637 | 11·5325626 | 5·1044687 | ·007518797 |
| 134 | 17956 | 2406104 | 11·5758369 | 5·1172299 | ·007462687 |
| 135 | 18225 | 2460375 | 11·6189500 | 5·1299278 | ·007407407 |
| 136 | 18496 | 2515456 | 11·6619038 | 5·1425632 | ·007352941 |
| 137 | 18769 | 2571353 | 11·7046999 | 5·1551367 | ·007299270 |
| 138 | 19044 | 2628072 | 11·7473401 | 5·1676493 | ·007246377 |
| 139 | 19321 | 2685619 | 11·7898261 | 5·1801015 | ·007194245 |
| 140 | 19600 | 2744000 | 11·8321596 | 5·1924941 | ·007142857 |
| 141 | 19881 | 2803221 | 11·8743422 | 5·2048279 | ·007092199 |
| 142 | 20164 | 2863288 | 11·9163753 | 5·2171034 | ·007042254 |
| 143 | 20449 | 2924207 | 11·9582607 | 5·2293215 | ·006993007 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 144 | 20736 | 2985984 | 12·0000000 | 5·2414828 | ·006944444 |
| 145 | 21025 | 3048625 | 12·0415946 | 5·2535879 | ·006896552 |
| 146 | 21316 | 3112136 | 12·0830460 | 5·2656374 | ·006849315 |
| 147 | 21609 | 3176523 | 12·1243557 | 5·2776321 | ·006802721 |
| 148 | 21904 | 3241792 | 12·1655251 | 5·2895725 | ·006756757 |
| 149 | 22201 | 3307949 | 12·2065556 | 5·3014592 | ·006711409 |
| 150 | 22500 | 3375000 | 12·2474487 | 5·3132928 | ·006666667 |
| 151 | 22801 | 3442951 | 12·2882057 | 5·3250740 | ·006622517 |
| 152 | 23104 | 3511808 | 12·3288280 | 5·3368033 | ·006578947 |
| 153 | 23409 | 3581577 | 12·3693169 | 5·3484812 | ·006535948 |
| 154 | 23716 | 3652264 | 12·4096736 | 5·3601084 | ·006493506 |
| 155 | 24025 | 3723875 | 12·4498996 | 5·3716854 | ·006451613 |
| 156 | 24336 | 3796416 | 12·4899960 | 5·3832126 | ·006410256 |
| 157 | 24649 | 3869893 | 12·5299641 | 5·3946907 | ·006369427 |
| 158 | 24964 | 3944312 | 12·5698051 | 5·4061202 | ·006329114 |
| 159 | 25281 | 4019679 | 12·6095202 | 5·4175015 | ·006289308 |
| 160 | 25600 | 4096000 | 12·6491106 | 5·4288352 | ·006250000 |
| 161 | 25921 | 4173281 | 12·6885775 | 5·4401218 | ·006211180 |
| 162 | 26244 | 4251528 | 12·7279221 | 5·4513618 | ·006172840 |
| 163 | 26569 | 4330747 | 12·7671453 | 5·4625556 | ·006134969 |
| 164 | 26896 | 4410944 | 12·8062485 | 5·4737037 | ·006097561 |
| 165 | 27225 | 4492125 | 12·8452326 | 5·4848066 | ·006060606 |
| 166 | 27556 | 4574296 | 12·8840987 | 5·4958647 | ·006024096 |
| 167 | 27889 | 4657463 | 12·9228480 | 5·5068784 | ·005988024 |
| 168 | 28224 | 4741632 | 12·9614814 | 5·5178484 | ·005952381 |
| 169 | 28561 | 4826809 | 13·0000000 | 5·5287748 | ·005917160 |
| 170 | 28900 | 4913000 | 13·0384048 | 5·5396583 | ·005882353 |
| 171 | 29241 | 5000211 | 13·0766968 | 5·5504991 | ·005847953 |
| 172 | 29584 | 5088448 | 13·1148770 | 5·5612978 | ·005813953 |
| 173 | 29929 | 5177717 | 13·1529464 | 5·5720546 | ·005780347 |
| 174 | 30276 | 5268024 | 13·1909060 | 5·5827702 | ·005747126 |
| 175 | 30625 | 5359375 | 13·2287566 | 5·5934447 | ·005714286 |
| 176 | 30976 | 5451776 | 13·2664992 | 5·6040787 | ·005681818 |
| 177 | 31329 | 5545233 | 13·3041347 | 5·6146724 | ·005649718 |
| 178 | 31684 | 5639752 | 13·3416641 | 5·6252263 | ·005617978 |
| 179 | 32041 | 5735339 | 13·3790882 | 5·6357408 | ·005586592 |
| 180 | 32400 | 5832000 | 13·4164079 | 5·6462162 | ·005555556 |
| 181 | 32761 | 5929741 | 13·4536240 | 5·6566528 | ·005524862 |
| 182 | 33124 | 6028568 | 13·4907376 | 5·6670511 | ·005494505 |
| 183 | 33489 | 6128487 | 13·5277493 | 5·6774114 | ·005464481 |
| 184 | 33856 | 6229504 | 13·5646600 | 5·6877340 | ·005434783 |
| 185 | 34225 | 6331625 | 13·6014705 | 5·6980192 | ·005405405 |
| 186 | 34596 | 6434856 | 13·6381817 | 5·7082675 | ·005376344 |
| 187 | 34969 | 6539203 | 13·6747943 | 5·7184791 | ·005347594 |
| 188 | 35344 | 6644672 | 13·7113092 | 5·7286543 | ·005319149 |
| 189 | 35721 | 6751269 | 13·7477271 | 5·7387936 | ·005291005 |
| 190 | 36100 | 6859000 | 13·7840488 | 5·7488971 | ·005263158 |
| 191 | 36481 | 6967871 | 13·8202750 | 5·7589652 | ·005235602 |
| 192 | 36864 | 7077888 | 13·8564065 | 5·7689982 | ·005208333 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 193 | 37249 | 7189057 | 13·8924440 | 5·7789966 | ·005181347 |
| 194 | 37636 | 7301384 | 13·9283883 | 5·7889604 | ·005154639 |
| 195 | 38025 | 7414875 | 13·9642400 | 5·7988900 | ·005128205 |
| 196 | 38416 | 7529536 | 14·0000000 | 5·8087857 | ·005102041 |
| 197 | 38809 | 7645373 | 14·0356688 | 5·8186479 | ·005076142 |
| 198 | 39204 | 7762392 | 14·0712473 | 5·8284767 | ·005050505 |
| 199 | 39601 | 7880599 | 14·1067360 | 5·8382725 | ·005025126 |
| 200 | 40000 | 8000000 | 14·1421356 | 5·8480355 | ·005000000 |
| 201 | 40401 | 8120601 | 14·1774469 | 5·8577660 | ·004975124 |
| 202 | 40804 | 8242408 | 14·2126704 | 5·8674643 | ·004950495 |
| 203 | 41209 | 8365427 | 14·2478068 | 5·8771307 | ·004926108 |
| 204 | 41616 | 8489664 | 14·2828569 | 5·8867653 | ·004901961 |
| 205 | 42025 | 8615125 | 14·3178211 | 5·8963685 | ·004878049 |
| 206 | 42436 | 8741816 | 14·3527001 | 5·9059406 | ·004854369 |
| 207 | 42849 | 8869743 | 14·3874946 | 5·9154817 | ·004830918 |
| 208 | 43264 | 8998912 | 14·4222051 | 5·9249921 | ·004807692 |
| 209 | 43681 | 9129329 | 14·4568323 | 5·9344721 | ·004784689 |
| 210 | 44100 | 9261000 | 14·4913767 | 5·9439220 | ·004761905 |
| 211 | 44521 | 9393931 | 14·5258390 | 5·9533418 | ·004739336 |
| 212 | 44944 | 9528128 | 14·5602198 | 5·9627320 | ·004716981 |
| 213 | 45369 | 9663597 | 14·5945195 | 5·9720926 | ·004694836 |
| 214 | 45796 | 9800344 | 14·6287388 | 5·9814240 | ·004672897 |
| 215 | 46225 | 9938375 | 14·6628783 | 5·9907264 | ·004651163 |
| 216 | 46656 | 10077696 | 14·6969385 | 6·0000000 | ·004629630 |
| 217 | 47089 | 10218313 | 14·7309199 | 6·0092450 | ·004608295 |
| 218 | 47524 | 10360232 | 14·7648231 | 6·0184617 | ·004587156 |
| 219 | 47961 | 10503459 | 14·7986486 | 6·0276502 | ·004566210 |
| 220 | 48400 | 10648000 | 14·8323970 | 6·0368107 | ·004545455 |
| 221 | 48841 | 10793861 | 14·8660687 | 6·0459435 | ·004524887 |
| 222 | 49284 | 10941048 | 14·8996644 | 6·0550489 | ·004504505 |
| 223 | 49729 | 11089567 | 14·9331845 | 6·0641270 | ·004484305 |
| 224 | 50176 | 11239424 | 14·9666295 | 6·0731779 | ·004464286 |
| 225 | 50625 | 11390625 | 15·0000000 | 6·0822020 | ·004444444 |
| 226 | 51076 | 11543176 | 15·0332964 | 6·0911994 | ·004424779 |
| 227 | 51529 | 11697083 | 15·0665192 | 6·1001702 | ·004405286 |
| 228 | 51984 | 11852352 | 15·0996689 | 6·1091147 | ·004385965 |
| 229 | 52441 | 12008989 | 15·1327460 | 6·1180332 | ·004366812 |
| 230 | 52900 | 12167000 | 15·1657509 | 6·1269257 | ·004347826 |
| 231 | 53361 | 12326391 | 15·1986842 | 6·1357924 | ·004329004 |
| 232 | 53824 | 12487168 | 15·2315462 | 6·1446337 | ·004310345 |
| 233 | 54289 | 12649337 | 15·2643375 | 6·1534495 | ·004291845 |
| 234 | 54756 | 12812904 | 15·2970585 | 6·1622401 | ·004273504 |
| 235 | 55225 | 12977875 | 15·3297097 | 6·1710058 | ·004255319 |
| 236 | 55696 | 13144256 | 15·3622915 | 6·1797466 | ·004237288 |
| 237 | 56169 | 13312053 | 15·3948043 | 6·1884628 | ·004219409 |
| 238 | 56644 | 13481272 | 15·4272486 | 6·1971544 | ·004201681 |
| 239 | 57121 | 13651919 | 15·4596248 | 6·2058218 | ·004184100 |
| 240 | 57600 | 13824000 | 15·4919334 | 6·2144650 | ·004166667 |
| 241 | 58081 | 13997521 | 15·5241747 | 6·2230843 | ·004149378 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 242 | 58564 | 14172488 | 15·5563492 | 6·2316797 | ·004132231 |
| 243 | 59049 | 14348907 | 15·5884573 | 6·2402515 | ·004115226 |
| 244 | 59536 | 14526784 | 15·6204994 | 6·2487998 | ·004098361 |
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| 257 | 66049 | 16974593 | 16·0312195 | 6·3578611 | ·003891051 |
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| 260 | 67600 | 17576000 | 16·1245155 | 6·3825043 | ·003846154 |
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| 262 | 68644 | 17984728 | 16·1864141 | 6·3988279 | ·003816794 |
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| 282 | 79524 | 22425768 | 16·7928556 | 6·5576722 | ·003546099 |
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| 316 | 99856 | 31554496 | 17·7763888 | 6·8112847 | ·003164557 |
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| 345 | 119025 | 41063625 | 18·5741756 | 7·0135791 | ·002898551 |
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| 365 | 133225 | 48627125 | 19·1049732 | 7·1465695 | ·002739726 |
| 366 | 133956 | 49027896 | 19·1311265 | 7·1530901 | ·002732240 |
| 367 | 134689 | 49430863 | 19·1572441 | 7·1595988 | ·002724796 |
| 368 | 135424 | 49836032 | 19·1833261 | 7·1660957 | ·002717391 |
| 369 | 136161 | 50243409 | 19·2093727 | 7·1725809 | ·002710027 |
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| 378 | 142884 | 54010152 | 19·4422221 | 7·2304268 | ·002645503 |
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| 445 | 198025 | 88121125 | 21·0950231 | 7·6346067 | ·002247191 |
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| 507 | 257049 | 130323843 | 22·5166605 | 7·9738731 | ·001972387 |
| 508 | 258064 | 131096512 | 22·5388553 | 7·9791122 | ·001968504 |
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| 559 | 312481 | 174676879 | 23·6431808 | 8·2376614 | ·001788909 |
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| 590 | 348100 | 205379000 | 24·2899156 | 8·3872065 | ·001694915 |
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| 592 | 350464 | 207474688 | 24·3310501 | 8·3966729 | ·001689189 |
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| 594 | 352836 | 209584584 | 24·3721152 | 8·4061180 | ·001683502 |
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| 597 | 356409 | 212776173 | 24·4335834 | 8·4202460 | ·001675042 |
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| 607 | 368449 | 223648543 | 24·6373700 | 8·4670000 | ·001647446 |
| 608 | 369664 | 224755712 | 24·6576560 | 8·4716471 | ·001644737 |
| 609 | 370881 | 225866529 | 24·6779254 | 8·4762892 | ·001642036 |
| 610 | 372100 | 226981000 | 24·6981781 | 8·4809261 | ·001639344 |
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| 613 | 375769 | 230346397 | 24·7588368 | 8·4948065 | ·001631321 |
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| 615 | 378225 | 232608375 | 24·7991935 | 8·5040350 | ·001626016 |
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| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
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| 1756 | 3083536 | 5414689216 | 41·9046537 | 12·0644679 | ·0005694761 |
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| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
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| 1765 | 3115225 | 5498372125 | 42·0119031 | 12·0850439 | ·0005665722 |
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| 1778 | 3161284 | 5620762952 | 42·1663373 | 12·1146420 | ·0005624297 |
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| 1787 | 3193369 | 5706550403 | 42·2729227 | 12·1350485 | ·0005595971 |
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| 1794 | 3218436 | 5773874184 | 42·3556371 | 12·1508729 | ·0005574136 |
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| 1797 | 3229209 | 5802888573 | 42·3910368 | 12·1576422 | ·0005564830 |
| 1798 | 3232804 | 5812581592 | 42·4028301 | 12·1598970 | ·0005561735 |
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| 1801 | 3243601 | 5841725401 | 42·4381903 | 12·1666562 | ·0005552471 |
| 1802 | 3247204 | 5851461608 | 42·4499705 | 12·1689076 | ·0005549390 |
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| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
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| 1832 | 3356224 | 6148602368 | 42·8018691 | 12·2360663 | ·0005458515 |
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| 1834 | 3363556 | 6168761704 | 42·8252262 | 12·2405174 | ·0005452563 |
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| 1861 | 3463321 | 6445240381 | 43·1393092 | 12·3002930 | ·0005373455 |
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| 1863 | 3470769 | 6466042647 | 43·1624837 | 12·3046978 | ·0005367687 |
| 1864 | 3474496 | 6476460544 | 43·1740663 | 12·3068990 | ·0005364807 |
| 1865 | 3478225 | 6486889625 | 43·1856458 | 12·3090994 | ·0005361930 |
| 1866 | 3481956 | 6497329896 | 43·1972221 | 12·3112991 | ·0005359057 |
| 1867 | 3485689 | 6507781363 | 43·2087954 | 12·3134979 | ·0005356186 |
| 1868 | 3489424 | 6518244032 | 43·2203656 | 12·3156959 | ·0005353319 |
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| 1870 | 3496900 | 6539203000 | 43·2434966 | 12·3200897 | ·0005347594 |
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| 1881 | 3538161 | 6655280841 | 43·3704969 | 12·3441995 | ·0005316321 |
| 1882 | 3541924 | 6665900968 | 43·3820239 | 12·3463866 | ·0005313496 |
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| 1884 | 3549456 | 6687175104 | 43·4050688 | 12·3507586 | ·0005307856 |
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| 1901 | 3613801 | 6869835701 | 43·6004587 | 12·3877959 | ·0005260389 |
| 1902 | 3617604 | 6880682808 | 43·6119249 | 12·3899676 | ·0005257624 |
| 1903 | 3621409 | 6891541327 | 43·6233882 | 12·3921386 | ·0005254861 |
| 1904 | 3625216 | 6902411264 | 43·6348485 | 12·3943089 | ·0005252101 |
| 1905 | 3629025 | 6913292625 | 43·6463057 | 12·3964784 | ·0005249344 |
| 1906 | 3632836 | 6924185416 | 43·6577599 | 12·3986471 | ·0005246590 |
| 1907 | 3636649 | 6935089643 | 43·6692111 | 12·4008151 | ·0005243838 |

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| 1910 | 3648100 | 6967871000 | 43·7035467 | 12·4073145 | ·0005235602 |
| 1911 | 3651921 | 6978821031 | 43·7149860 | 12·4094794 | ·0005232862 |
| 1912 | 3655744 | 6989782528 | 43·7264222 | 12·4116436 | ·0005230126 |
| 1913 | 3659569 | 7000755497 | 43·7378554 | 12·4138070 | ·0005227392 |
| 1914 | 3663396 | 7011739944 | 43·7492857 | 12·4159697 | ·0005224660 |
| 1915 | 3667225 | 7022735875 | 43·7607129 | 12·4181316 | ·0005221932 |
| 1916 | 3671056 | 7033743296 | 43·7721373 | 12·4202928 | ·0005219207 |
| 1917 | 3674889 | 7044762213 | 43·7835585 | 12·4224533 | ·0005216484 |
| 1918 | 3678724 | 7055792632 | 43·7949768 | 12·4246129 | ·0005213764 |
| 1919 | 3682561 | 7066834559 | 43·8063922 | 12·4267719 | ·0005211047 |
| 1920 | 3686400 | 7077888000 | 43·8178046 | 12·4289300 | ·0005208333 |
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| 1922 | 3694084 | 7100029448 | 43·8406204 | 12·4332441 | ·0005202914 |
| 1923 | 3697929 | 7111117467 | 43·8520239 | 12·4354001 | ·0005200208 |
| 1924 | 3701776 | 7122217024 | 43·8634244 | 12·4375552 | ·0005197505 |
| 1925 | 3705625 | 7133328125 | 43·8748219 | 12·4397097 | ·0005194805 |
| 1926 | 3709476 | 7144450776 | 43·8862165 | 12·4418634 | ·0005192108 |
| 1927 | 3713329 | 7155584983 | 43·8976081 | 12·4440163 | ·0005189414 |
| 1928 | 3717184 | 7166730752 | 43·9089968 | 12·4461685 | ·0005186722 |
| 1929 | 3721041 | 7177888089 | 43·9203725 | 12·4483200 | ·0005184033 |
| 1930 | 3724900 | 7189057000 | 43·9317652 | 12·4504707 | ·0005181347 |
| 1931 | 3728761 | 7200237491 | 43·9431451 | 12·4526206 | ·0005178664 |
| 1932 | 3732624 | 7211429568 | 43·9545220 | 12·4547699 | ·0005175983 |
| 1933 | 3736489 | 7222633237 | 43·9658959 | 12·4569184 | ·0005173306 |
| 1934 | 3740356 | 7233848504 | 43·9772668 | 12·4590661 | ·0005170631 |
| 1935 | 3744225 | 7245075375 | 43·9886349 | 12·4612131 | ·0005167959 |
| 1936 | 3748096 | 7256313856 | 44·0000000 | 12·4633594 | ·0005165289 |
| 1937 | 3751969 | 7267563953 | 44·0113622 | 12·4655049 | ·0005162623 |
| 1938 | 3755844 | 7278825672 | 44·0227214 | 12·4676497 | ·0005159959 |
| 1939 | 3759721 | 7290099019 | 44·0340777 | 12·4697937 | ·0005157298 |
| 1940 | 3763600 | 7301384000 | 44·0454311 | 12·4719370 | ·0005154639 |
| 1941 | 3767481 | 7312680621 | 44·0567815 | 12·4740796 | ·0005151984 |
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| 1953 | 3814209 | 7449150177 | 44·1027596 | 12·4997333 | ·0005120328 |
| 1954 | 3818116 | 7460598664 | 44·2040722 | 12·5018664 | ·0005117707 |
| 1955 | 3822025 | 7472058875 | 44·2153819 | 12·5039988 | ·0005115090 |
| 1956 | 3825936 | 7483530816 | 44·2266888 | 12·5061304 | ·0005112474 |

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| 1958 | 3833764 | 7506509912 | 44·2492938 | 12·5103914 | ·0005107252 |
| 1959 | 3837681 | 7518017079 | 44·2605919 | 12·5125208 | ·0005104645 |
| 1960 | 3841600 | 7529536000 | 44·2718872 | 12·5146495 | ·0005102041 |
| 1961 | 3845521 | 7541066681 | 44·2831797 | 12·5167775 | ·0005099439 |
| 1962 | 3849444 | 7552609128 | 44·2944692 | 12·5189047 | ·0005096840 |
| 1963 | 3853369 | 7564163347 | 44·3057558 | 12·5210313 | ·0005094244 |
| 1964 | 3857296 | 7575729344 | 44·3170396 | 12·5231571 | ·0005091650 |
| 1965 | 3861225 | 7587307125 | 44·3283205 | 12·5252822 | ·0005089059 |
| 1966 | 3865156 | 7598896696 | 44·3395985 | 12·5274065 | ·0005086470 |
| 1967 | 3869089 | 7610498063 | 44·3508737 | 12·5295302 | ·0005083884 |
| 1968 | 3873024 | 7622111232 | 44·3621460 | 12·5316531 | ·0005081301 |
| 1969 | 3876961 | 7633736209 | 44·3734155 | 12·5337753 | ·0005078720 |
| 1970 | 3880900 | 7645373000 | 44·3846820 | 12·5358968 | ·0005076142 |
| 1971 | 3884841 | 7657021611 | 44·3959457 | 12·5380176 | ·0005073567 |
| 1972 | 3888784 | 7668682048 | 44·4072066 | 12·5401377 | ·0005070994 |
| 1973 | 3892729 | 7680354317 | 44·4184646 | 12·5422570 | ·0005068424 |
| 1974 | 3896676 | 7692038424 | 44·4297198 | 12·5443757 | ·0005065856 |
| 1975 | 3900625 | 7703734375 | 44·4409720 | 12·5464936 | ·0005063291 |
| 1976 | 3904576 | 7715442176 | 44·4522215 | 12·5486107 | ·0005060729 |
| 1977 | 3908529 | 7727161833 | 44·4634681 | 12·5507272 | ·0005058169 |
| 1978 | 3912484 | 7738893352 | 44·4747119 | 12·5528430 | ·0005055612 |
| 1979 | 3916441 | 7750636739 | 44·4859528 | 12·5549580 | ·0005053057 |
| 1980 | 3920400 | 7762392000 | 44·4971909 | 12·5570723 | ·0005050505 |
| 1981 | 3924361 | 7774159141 | 44·5084262 | 12·5591860 | ·0005047956 |
| 1982 | 3928324 | 7785938168 | 44·5196586 | 12·5612989 | ·0005045409 |
| 1983 | 3932289 | 7797729087 | 44·5308881 | 12·5634111 | ·0005042864 |
| 1984 | 3936256 | 7809531904 | 44·5421149 | 12·5655226 | ·0005040323 |
| 1985 | 3940225 | 7821346625 | 44·5533388 | 12·5676334 | ·0005037783 |
| 1986 | 3944196 | 7833173256 | 44·5645599 | 12·5697435 | ·0005035247 |
| 1987 | 3948169 | 7845011803 | 44·5757781 | 12·5718529 | ·0005032713 |
| 1988 | 3952144 | 7856862272 | 44·5869936 | 12·5739615 | ·0005030181 |
| 1989 | 3956121 | 7868724669 | 44·5982062 | 12·5760695 | ·0005027652 |
| 1990 | 3960100 | 7880599000 | 44·6094160 | 12·5781767 | ·0005025126 |
| 1991 | 3964081 | 7892485271 | 44·6206230 | 12·5802832 | ·0005022602 |
| 1992 | 3968064 | 7904383488 | 44·6318272 | 12·5823891 | ·0005020080 |
| 1993 | 3972049 | 7916293657 | 44·6430286 | 12·5844942 | ·0005017561 |
| 1994 | 3976036 | 7928215784 | 44·6542271 | 12·5865987 | ·0005015045 |
| 1995 | 3980025 | 7940149875 | 44·6654228 | 12·5887024 | ·0005012531 |
| 1996 | 3984016 | 7952095936 | 44·6766158 | 12·5908054 | ·0005010020 |
| 1997 | 3988009 | 7964053973 | 44·6878059 | 12·5929078 | ·0005007511 |
| 1998 | 3992004 | 7976023992 | 44·6989933 | 12·5950094 | ·0005005005 |
| 1999 | 3996001 | 7988005999 | 44·7101778 | 12·5971103 | ·0005002501 |
| 2000 | 4000000 | 8000000000 | 44·7213596 | 12·5992105 | ·0005000000 |
| 2001 | 4004001 | 8012006001 | 44·7325385 | 12·6013101 | ·0004997501 |
| 2002 | 4008004 | 8024024008 | 44·7437146 | 12·6034089 | ·0004995005 |
| 2003 | 4012009 | 8036054027 | 44·7548880 | 12·6055070 | ·0004992511 |
| 2004 | 4016016 | 8048096064 | 44·7660586 | 12·6076044 | ·0004990020 |
| 2005 | 4020025 | 8060150125 | 44·7772264 | 12·6097011 | ·0004987531 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 2006 | 4024036 | 8072216216 | 44·7883913 | 12·6117971 | ·0004985045 |
| 2007 | 4028049 | 8084294343 | 44·7995535 | 12·6138924 | ·0004982561 |
| 2008 | 4032064 | 8096384512 | 44·8107130 | 12·6159870 | ·0004980080 |
| 2009 | 4036081 | 8108486729 | 44·8218697 | 12·6180810 | ·0004977601 |
| 2010 | 4040100 | 8120601000 | 44·8330235 | 12·6201743 | ·0004975124 |
| 2011 | 4044121 | 8132727331 | 44·8441746 | 12·6222669 | ·0004972650 |
| 2012 | 4048144 | 8144865728 | 44·8553230 | 12·6243587 | ·0004970179 |
| 2013 | 4052169 | 8157016197 | 44·8664685 | 12·6264499 | ·0004967710 |
| 2014 | 4056196 | 8169178744 | 44·8776113 | 12·6285404 | ·0004965243 |
| 2015 | 4060225 | 8181353375 | 44·8887514 | 12·6306301 | ·0004962779 |
| 2016 | 4064256 | 8193540096 | 44·8998886 | 12·6327192 | ·0004060317 |
| 2017 | 4068289 | 8205738913 | 44·9110231 | 12·6348076 | ·0004957858 |
| 2018 | 4072324 | 8217949832 | 44·9221549 | 12·6368953 | ·0004955401 |
| 2019 | 4076361 | 8230172859 | 44·9332839 | 12·6389823 | ·0004952947 |
| 2020 | 4080400 | 8242408000 | 44·9444101 | 12·6410687 | ·0004950495 |
| 2021 | 4084441 | 8254655261 | 44·9555336 | 12·6431543 | ·0004948046 |
| 2022 | 4088484 | 8266914648 | 44·9666543 | 12·6452393 | ·0004945598 |
| 2023 | 4092529 | 8279186167 | 44·9777723 | 12·6473235 | ·0004943154 |
| 2024 | 4096576 | 8291469824 | 44·9888875 | 12·6494071 | ·0004940711 |
| 2025 | 4100625 | 8303765625 | 45·0000000 | 12·6514900 | ·0004938272 |
| 2026 | 4104676 | 8316073576 | 45·0111097 | 12·6535722 | ·0004935834 |
| 2027 | 4108729 | 8328393683 | 45·0222167 | 12·6556538 | ·0004933399 |
| 2028 | 4112784 | 8340725952 | 45·0333210 | 12·6577346 | ·0004930966 |
| 2029 | 4116841 | 8353070389 | 45·0444225 | 12·6598148 | ·0004928536 |
| 2030 | 4120900 | 8365427000 | 45·0555213 | 12·6618943 | ·0004926108 |
| 2031 | 4124961 | 8377795791 | 45·0666173 | 12·6639731 | ·0004923683 |
| 2032 | 4129024 | 8390176768 | 45·0777107 | 12·6660512 | ·0004921260 |
| 2033 | 4133089 | 8402569937 | 45·0888013 | 12·6681286 | ·0004918839 |
| 2034 | 4137156 | 8414975304 | 45·0998891 | 12·6702053 | ·0004916421 |
| 2035 | 4141225 | 8427392875 | 45·1109743 | 12·6722814 | ·0004914005 |
| 2036 | 4145296 | 8439822656 | 45·1220567 | 12·6743567 | ·0004911591 |
| 2037 | 4149369 | 8452264653 | 45·1331364 | 12·6764314 | ·0004909180 |
| 2038 | 4153444 | 8464718872 | 45·1442134 | 12·6785054 | ·0004906771 |
| 2039 | 4157521 | 8477185319 | 45·1552876 | 12·6805788 | ·0004904365 |
| 2040 | 4161600 | 8489664000 | 45·1663592 | 12·6826514 | ·0004901961 |
| 2041 | 4165681 | 8502154921 | 45·1774280 | 12·6847234 | ·0004899559 |
| 2042 | 4169764 | 8514658088 | 45·1884941 | 12·6867947 | ·0004897160 |
| 2043 | 4173849 | 8527173507 | 45·1995575 | 12·6888654 | ·0004894762 |
| 2044 | 4177936 | 8539701184 | 45·2106182 | 12·6909354 | ·0004892368 |
| 2045 | 4182025 | 8552241125 | 45·2216762 | 12·6930047 | ·0004889976 |
| 2046 | 4186116 | 8564793336 | 45·2327315 | 12·6950733 | ·0004887586 |
| 2047 | 4190209 | 8577357823 | 45·2437841 | 12·6971412 | ·0004885198 |
| 2048 | 4194304 | 8589934592 | 45·2548340 | 12·6992084 | ·0004882813 |
| 2049 | 4198401 | 8602523649 | 45·2658812 | 12·7012750 | ·0004880429 |
| 2050 | 4202500 | 8615125000 | 45·2769257 | 12·7033409 | ·0004878049 |
| 2051 | 4206601 | 8627738651 | 45·2879675 | 12·7054061 | ·0004875670 |
| 2052 | 4210704 | 8640364608 | 45·2990066 | 12·7074707 | ·0004873294 |
| 2053 | 4214809 | 8653002877 | 45·3100430 | 12·7095346 | ·0004870921 |
| 2054 | 4218916 | 8665653464 | 45·3210768 | 12·7115978 | ·0004868549 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 2055 | 4223025 | 8678316375 | 45·3321078 | 12·7136603 | ·0004866180 |
| 2056 | 4227136 | 8690991616 | 45·3431362 | 12·7157222 | ·0004863813 |
| 2057 | 4231249 | 8703679193 | 45·3541619 | 12·7177835 | ·0004861449 |
| 2058 | 4235364 | 8716379112 | 45·3651849 | 12·7198441 | ·0004859086 |
| 2059 | 4239481 | 8729091379 | 45·3762052 | 12·7219040 | ·0004856727 |
| 2060 | 4243600 | 8741816000 | 45·3872229 | 12·7239632 | ·0004854369 |
| 2061 | 4247721 | 8754552981 | 45·3982378 | 12·7260218 | ·0004852014 |
| 2062 | 4251844 | 8767302328 | 45·4092501 | 12·7280797 | ·0004849661 |
| 2063 | 4255969 | 8780064047 | 45·4202598 | 12·7301370 | ·0004847310 |
| 2064 | 4260096 | 8792838144 | 45·4312668 | 12·7321935 | ·0004844961 |
| 2065 | 4264225 | 8805624625 | 45·4422711 | 12·7342494 | ·0004842615 |
| 2066 | 4268356 | 8818423496 | 45·4532727 | 12·7363046 | ·0004840271 |
| 2067 | 4272489 | 8831234763 | 45·4642717 | 12·7383592 | ·0004837929 |
| 2068 | 4276624 | 8844058432 | 45·4752680 | 12·7404131 | ·0004835590 |
| 2069 | 4280761 | 8856894509 | 45·4862616 | 12·7424664 | ·0004833253 |
| 2070 | 4284900 | 8869743000 | 45·4972526 | 12·7445189 | ·0004830918 |
| 2071 | 4289041 | 8882603911 | 45·5082410 | 12·7465709 | ·0004828585 |
| 2072 | 4293184 | 8895477248 | 45·5192267 | 12·7486222 | ·0004826255 |
| 2073 | 4297329 | 8908363017 | 45·5302097 | 12·7506728 | ·0004823927 |
| 2074 | 4301476 | 8921261224 | 45·5411901 | 12·7527227 | ·0004821601 |
| 2075 | 4305625 | 8934171875 | 45·5521679 | 12·7547721 | ·0004819277 |
| 2076 | 4309776 | 8947094976 | 45·5631430 | 12·7568207 | ·0004816956 |
| 2077 | 4313929 | 8960030533 | 45·5741155 | 12·7588687 | ·0004814636 |
| 2078 | 4318084 | 8972978552 | 45·5850853 | 12·7609160 | ·0004812320 |
| 2079 | 4322241 | 8985939039 | 45·5960525 | 12·7629627 | ·0004810005 |
| 2080 | 4326400 | 8998912000 | 45·6070170 | 12·7650087 | ·0004807692 |
| 2081 | 4330561 | 9011897441 | 45·6179789 | 12·7670540 | ·0004805382 |
| 2082 | 4334724 | 9024895368 | 45·6289382 | 12·7690987 | ·0004803074 |
| 2083 | 4338889 | 9037905787 | 45·6398948 | 12·7711427 | ·0004800768 |
| 2084 | 4343056 | 9050928704 | 45·6508488 | 12·7731861 | ·0004798464 |
| 2085 | 4347225 | 9063964125 | 45·6618002 | 12·7752288 | ·0004796163 |
| 2086 | 4351396 | 9077012056 | 45·6727490 | 12·7772709 | ·0004793864 |
| 2087 | 4355569 | 9090072503 | 45·6836951 | 12·7793123 | ·0004791567 |
| 2088 | 4359744 | 9103145472 | 45·6946386 | 12·7813531 | ·0004789272 |
| 2089 | 4363921 | 9116230969 | 45·7055795 | 12·7833932 | ·0004786979 |
| 2090 | 4368100 | 9129329000 | 45·7165178 | 12·7854326 | ·0004784689 |
| 2091 | 4372281 | 9142439571 | 45·7274534 | 12·7874714 | ·0004782401 |
| 2092 | 4376464 | 9155562688 | 45·7383865 | 12·7895096 | ·0004780115 |
| 2093 | 4380649 | 9168698357 | 45·7493169 | 12·7915471 | ·0004777831 |
| 2094 | 4384836 | 9181846584 | 45·7602447 | 12·7935840 | ·0004775549 |
| 2095 | 4389025 | 9195007375 | 45·7711699 | 12·7956202 | ·0004773270 |
| 2096 | 4393216 | 9208180736 | 45·7820926 | 12·7976558 | ·0004770992 |
| 2097 | 4397409 | 9221366673 | 45·7930126 | 12·7996907 | ·0004768717 |
| 2098 | 4401604 | 9234565192 | 45·8039299 | 12·8017250 | ·0004766444 |
| 2099 | 4405801 | 9247776299 | 45·8148447 | 12·8037586 | ·0004764173 |
| 2100 | 4410000 | 9261000000 | 45·8257569 | 12·8057916 | ·0004761905 |
| 2101 | 4414201 | 9274236301 | 45·8366665 | 12·8078239 | ·0004759638 |
| 2102 | 4418404 | 9287485208 | 45·8475735 | 12·8098556 | ·0004757374 |
| 2103 | 4422609 | 9300746727 | 45·8584779 | 12·8118866 | ·0004755112 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 2104 | 4426816 | 9314020864 | 45·8693798 | 12·8139170 | ·0004752852 |
| 2105 | 4431025 | 9327307625 | 45·8802790 | 12·8159468 | ·0004750594 |
| 2106 | 4435236 | 9340607016 | 45·8911756 | 12·8179759 | ·0004748338 |
| 2107 | 4439449 | 9353919043 | 45·9020696 | 12·8200044 | ·0004746084 |
| 2108 | 4443664 | 9367243712 | 45·9129611 | 12·8220323 | ·0004743833 |
| 2109 | 4447881 | 9380581029 | 45·9238500 | 12·8240595 | ·0004741584 |
| 2110 | 4452100 | 9393931000 | 45·9347363 | 12·8260861 | ·0004739336 |
| 2111 | 4456321 | 9407293631 | 45·9456200 | 12·8281120 | ·0004737091 |
| 2112 | 4460544 | 9420668928 | 45·9565012 | 12·8301378 | ·0004734848 |
| 2113 | 4464769 | 9434056897 | 45·9673798 | 12·8321620 | ·0004732608 |
| 2114 | 4468996 | 9447457544 | 45·9782557 | 12·8341860 | ·0004730369 |
| 2115 | 4473225 | 9460870875 | 45·9891291 | 12·8362094 | ·0004728132 |
| 2116 | 4477456 | 9474296896 | 46·0000000 | 12·8382321 | ·0004725898 |
| 2117 | 4481689 | 9487735613 | 46·0108683 | 12·8402542 | ·0004723666 |
| 2118 | 4485924 | 9501187032 | 46·0217340 | 12·8422756 | ·0004721435 |
| 2119 | 4490161 | 9514651159 | 46·0325971 | 12·8442964 | ·0004719207 |
| 2120 | 4494400 | 9528128000 | 46·0434577 | 12·8463166 | ·0004716981 |
| 2121 | 4498641 | 9541617561 | 46·0543158 | 12·8483361 | ·0004714757 |
| 2122 | 4502884 | 9555119848 | 46·0651712 | 12·8503551 | ·0004712535 |
| 2123 | 4507129 | 9568634867 | 46·0760241 | 12·8523733 | ·0004710316 |
| 2124 | 4511376 | 9582162624 | 46·0868745 | 12·8543910 | ·0004708098 |
| 2125 | 4515625 | 9595703125 | 46·0977223 | 12·8564080 | ·0004705882 |
| 2126 | 4519876 | 9609256376 | 46·1085675 | 12·8584243 | ·0004703669 |
| 2127 | 4524129 | 9622822383 | 46·1194102 | 12·8604401 | ·0004701457 |
| 2128 | 4528384 | 9636401152 | 46·1302504 | 12·8624552 | ·0004699248 |
| 2129 | 4532641 | 9649992689 | 46·1410880 | 12·8644697 | ·0004697041 |
| 2130 | 4536900 | 9663597000 | 46·1519230 | 12·8664835 | ·0004694836 |
| 2131 | 4541161 | 9677214091 | 46·1627555 | 12·8684967 | ·0004692633 |
| 2132 | 4545424 | 9690843968 | 46·1735855 | 12·8705093 | ·0004690432 |
| 2133 | 4549689 | 9704486637 | 46·1844130 | 12·8725213 | ·0004688233 |
| 2134 | 4553956 | 9718142104 | 46·1952378 | 12·8745326 | ·0004686036 |
| 2135 | 4558225 | 9731810375 | 46·2060602 | 12·8765433 | ·0004683841 |
| 2136 | 4562496 | 9745491456 | 46·2168800 | 12·8785534 | ·0004681648 |
| 2137 | 4566769 | 9759185353 | 46·2276973 | 12·8805628 | ·0004679457 |
| 2138 | 4571044 | 9772892072 | 46·2385121 | 12·8825717 | ·0004677268 |
| 2139 | 4575321 | 9786611619 | 46·2493243 | 12·8845199 | ·0004675082 |
| 2140 | 4579600 | 9800344000 | 46·2601340 | 12·8865874 | ·0004672897 |
| 2141 | 4583881 | 9814089221 | 46·2709412 | 12·8885944 | ·0004670715 |
| 2142 | 4588164 | 9827847288 | 46·2817459 | 12·8906007 | ·0004668534 |
| 2143 | 4592449 | 9841618207 | 46·2925480 | 12·8926064 | ·0004666356 |
| 2144 | 4596736 | 9855401984 | 46·3033476 | 12·8946115 | ·0004664179 |
| 2145 | 4601025 | 9869198625 | 46·3141447 | 12·8966159 | ·0004662005 |
| 2146 | 4605316 | 9883008136 | 46·3249393 | 12·8986197 | ·0004659832 |
| 2147 | 4609609 | 9896830523 | 46·3357314 | 12·9006229 | ·0004657662 |
| 2148 | 4613904 | 9910665792 | 46·3465209 | 12·9026255 | ·0004655493 |
| 2149 | 4618201 | 9924513949 | 46·3573079 | 12·9046275 | ·0004653327 |
| 2150 | 4622500 | 9938375000 | 46·3680924 | 12·9066288 | ·0004651163 |
| 2151 | 4626801 | 9952248951 | 46·3788745 | 12·9086295 | ·0004649000 |
| 2152 | 4631104 | 9966135808 | 46·3896540 | 12·9106296 | ·0004646840 |

| No. | Square | Cube | Square Root | Cube Root | Reciprocal |
|---|---|---|---|---|---|
| 2153 | 4635409 | 9980035577 | 46·4004310 | 12·9126291 | ·0004644682 |
| 2154 | 4639716 | 9993948264 | 46·4112055 | 12·9146279 | ·0004642526 |
| 2155 | 4644025 | 10007873875 | 46·4219775 | 12·9166262 | ·0004640371 |
| 2156 | 4648336 | 10021812416 | 46·4327471 | 12·9186238 | ·0004638219 |
| 2157 | 4652649 | 10035763893 | 46·4435141 | 12·9206208 | ·0004636069 |
| 2158 | 4656964 | 10049728312 | 46·4542786 | 12·9226172 | ·0004633920 |
| 2159 | 4661281 | 10063705679 | 46·4650406 | 12·9246129 | ·0004631774 |
| 2160 | 4665600 | 10077696000 | 46·4758002 | 12·9266081 | ·0004629630 |
| 2161 | 4669921 | 10091699281 | 46·4865572 | 12·9286027 | ·0004627487 |
| 2162 | 4674244 | 10105715528 | 46·4973118 | 12·9305966 | ·0004625347 |
| 2163 | 4678569 | 10119744747 | 46·5080638 | 12·9325899 | ·0004623209 |
| 2164 | 4682896 | 10133786944 | 46·5188134 | 12·9345827 | ·0004621072 |
| 2165 | 4687225 | 10147842125 | 46·5295605 | 12·9365747 | ·0004618938 |
| 2166 | 4691556 | 10161910296 | 46·5403051 | 12·9385662 | ·0004616805 |
| 2167 | 4695889 | 10175991463 | 46·5510472 | 12·9405570 | ·0004614675 |
| 2168 | 4700224 | 10190085632 | 46·5617869 | 12·9425472 | ·0004612546 |
| 2169 | 4704561 | 10204192809 | 46·5725241 | 12·9445369 | ·0004610420 |
| 2170 | 4708900 | 10218313000 | 46·5832588 | 12·9465259 | ·0004608295 |
| 2171 | 4713241 | 10232446211 | 46·5939910 | 12·9485143 | ·0004606172 |
| 2172 | 4717584 | 10246592448 | 46·6047208 | 12·9505021 | ·0004604052 |
| 2173 | 4721929 | 10260751717 | 46·6154481 | 12·9524893 | ·0004601933 |
| 2174 | 4726276 | 10274924024 | 46·6261729 | 12·9544759 | ·0004599816 |
| 2175 | 4730625 | 10289109375 | 46·6368953 | 12·9564618 | ·0004597701 |
| 2176 | 4734976 | 10303307776 | 46·6476152 | 12·9584472 | ·0004595588 |
| 2177 | 4739329 | 10317519233 | 46·6583326 | 12·9604319 | ·0004593477 |
| 2178 | 4743684 | 10331743752 | 46·6690476 | 12·9624161 | ·0004591368 |
| 2179 | 4748041 | 10345981339 | 46·6797601 | 12·9643996 | ·0004589261 |
| 2180 | 4752400 | 10360232000 | 46·6904701 | 12·9663826 | ·0004587156 |
| 2181 | 4756761 | 10374495741 | 46·7011777 | 12·9683649 | ·0004585053 |
| 2182 | 4761124 | 10388772568 | 46·7118829 | 12·9703466 | ·0004582951 |
| 2183 | 4765489 | 10403062487 | 46·7225855 | 12·9723277 | ·0004580852 |
| 2184 | 4769856 | 10417365504 | 46·7332858 | 12·9743082 | ·0004578755 |
| 2185 | 4774225 | 10431681625 | 46·7439836 | 12·9762881 | ·0004576659 |
| 2186 | 4778596 | 10446010856 | 46·7546789 | 12·9782674 | ·0004574565 |
| 2187 | 4782969 | 10460353203 | 46·7653718 | 12·9802461 | ·0004572474 |
| 2188 | 4787344 | 10474708672 | 46·7760623 | 12·9822242 | ·0004570384 |
| 2189 | 4791721 | 10489077269 | 46·7867503 | 12·9842017 | ·0004568296 |
| 2190 | 4796100 | 10503459000 | 46·7974358 | 12·9861786 | ·0004566210 |
| 2191 | 4800481 | 10517853871 | 46·8081189 | 12·9881549 | ·0004564126 |
| 2192 | 4804864 | 10532261888 | 46·8187996 | 12·9901306 | ·0004562044 |
| 2193 | 4809249 | 10546683057 | 46·8294779 | 12·9921057 | ·0004559964 |
| 2194 | 4813636 | 10561117384 | 46·8401537 | 12·9940802 | ·0004557885 |
| 2195 | 4818025 | 10575564875 | 46·8508271 | 12·9960540 | ·0004555809 |
| 2196 | 4822416 | 10590025536 | 46·8614981 | 12·9980273 | ·0004553734 |
| 2197 | 4826809 | 10604499373 | 46·8721666 | 13·0000000 | ·0004551661 |
| 2198 | 4831204 | 10618986392 | 46·8828327 | 13·0019721 | ·0004549591 |
| 2199 | 4835601 | 10633486599 | 46·8934963 | 13·0039436 | ·0004547522 |
| 2200 | 4840000 | 10648000000 | 46·9041576 | 13·0059145 | ·0004545455 |
| 2201 | 4844401 | 10662526601 | 46·9148164 | 13·0078848 | ·0004543389 |

## Table CXCVII.—Fourth Powers of Numbers.

| No. | 4th Power. | No. | 4th Power. | No. | 4th Power. | No. | 4th Power. |
|---|---|---|---|---|---|---|---|
| 1 | 1 | 26 | 456,976 | 51 | 6,765,201 | 76 | 33,362,176 |
| 2 | 16 | 27 | 531,441 | 52 | 7,311,616 | 77 | 35,153,041 |
| 3 | 81 | 28 | 614,656 | 53 | 7,890,481 | 78 | 37,015,056 |
| 4 | 256 | 29 | 707,281 | 54 | 8,503,056 | 79 | 38,950,081 |
| 5 | 625 | 30 | 810,000 | 55 | 9,150,625 | 80 | 40,960,000 |
| 6 | 1,296 | 31 | 923,521 | 56 | 9,834,496 | 81 | 43,046,721 |
| 7 | 2,401 | 32 | 1,048,576 | 57 | 10,556,001 | 82 | 45,212,176 |
| 8 | 4,096 | 33 | 1,185,921 | 58 | 11,316,496 | 83 | 47,458,321 |
| 9 | 6,561 | 34 | 1,336,336 | 59 | 12,117,361 | 84 | 49,787,136 |
| 10 | 10,000 | 35 | 1,500,625 | 60 | 12,960,000 | 85 | 52,200,625 |
| 11 | 14,641 | 36 | 1,679,616 | 61 | 13,845,841 | 86 | 54,708,016 |
| 12 | 20,736 | 37 | 1,874,161 | 62 | 14,776,336 | 87 | 57,289,761 |
| 13 | 28,561 | 38 | 2,085,136 | 63 | 15,752,961 | 88 | 59,969,536 |
| 14 | 38,416 | 39 | 2,313,441 | 64 | 16,777,216 | 89 | 62,742,241 |
| 15 | 50,625 | 40 | 2,560,000 | 65 | 17,850,625 | 90 | 65,610,000 |
| 16 | 65,536 | 41 | 2,825,761 | 66 | 18,974,736 | 91 | 68,574,961 |
| 17 | 83,521 | 42 | 3,111,696 | 67 | 20,151,121 | 92 | 71,639,296 |
| 18 | 104,976 | 43 | 3,418,801 | 68 | 21,381,376 | 93 | 74,805,201 |
| 19 | 130,321 | 44 | 3,748,096 | 69 | 22,667,121 | 94 | 78,074,896 |
| 20 | 160,000 | 45 | 4,100,625 | 70 | 24,010,000 | 95 | 81,450,625 |
| 21 | 194,481 | 46 | 4,477,456 | 71 | 25,411,681 | 96 | 84,934,656 |
| 22 | 234,256 | 47 | 4,879,681 | 72 | 26,873,856 | 97 | 88,529,281 |
| 23 | 279,841 | 48 | 5,308,416 | 73 | 28,398,241 | 98 | 92,236,816 |
| 24 | 331,776 | 49 | 5,764,801 | 74 | 29,986,576 | 99 | 96,059,601 |
| 25 | 390,625 | 50 | 6,250,000 | 75 | 31,640,625 | 100 | 100,000,000 |

## Table CXCVIIa.—Fourth Root of Numbers.

| No. | 4th Root. | No. | 4th Root. | No. | 4th Root. | No. | 4th Root. |
|---|---|---|---|---|---|---|---|
| 1 | 1·000 | 13 | 1·899 | 25 | 2·236 | 37 | 2·466 |
| 2 | 1·189 | 14 | 1·934 | 26 | 2·258 | 38 | 2·482 |
| 3 | 1·316 | 15 | 1·968 | 27 | 2·279 | 39 | 2·499 |
| 4 | 1·414 | 16 | 2·000 | 28 | 2·300 | 40 | 2·515 |
| 5 | 1·495 | 17 | 2·031 | 29 | 2·320 | 41 | 2·530 |
| 6 | 1·565 | 18 | 2·060 | 30 | 2·340 | 42 | 2·546 |
| 7 | 1·626 | 19 | 2·088 | 31 | 2·360 | 43 | 2·560 |
| 8 | 1·682 | 20 | 2·115 | 32 | 2·378 | 44 | 2·574 |
| 9 | 1·732 | 21 | 2·141 | 33 | 2·397 | 45 | 2·590 |
| 10 | 1·778 | 22 | 2·166 | 34 | 2·415 | 46 | 2·605 |
| 11 | 1·821 | 23 | 2·190 | 35 | 2·432 | 47 | 2·618 |
| 12 | 1·861 | 24 | 2·215 | 36 | 2·449 | 48 | 2·631 |

## Table CXCVIIb.—Fourth Powers of Shaft Diameters.

| Dia. | 4th Power. | Dia. | 4th Power. | Dia. | 4th Power. | Dia. | 4th Power. | Dia. | 4th Power. |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1·00 | 4⅜ | 366 | 7¾ | 3,607 | 12 | 20,736 | 18½ | 117,160 |
| 1⅛ | 1·60 | 4½ | 410 | 7⅞ | 3,846 | 12¼ | 22,500 | 18¾ | 123,600 |
| 1¼ | 2·44 | 4⅝ | 458 | 8 | 4,096 | 12½ | 24,430 | 19 | 130,321 |
| 1⅜ | 3·57 | 4¾ | 509 | 8⅛ | 4,358 | 12¾ | 26,439 | 19¼ | 137,400 |
| 1½ | 5·06 | 4⅞ | 565 | 8¼ | 4,632 | 13 | 28,561 | 19½ | 144,600 |
| 1⅝ | 6·97 | 5 | 625 | 8⅜ | 4,920 | 13¼ | 30,835 | 19¾ | 152,100 |
| 1¾ | 9·38 | 5⅛ | 690 | 8½ | 5,220 | 13½ | 33,233 | 20 | 160,000 |
| 1⅞ | 12·4 | 5¼ | 760 | 8⅝ | 5,534 | 13¾ | 35,759 | 20¼ | 168,100 |
| 2 | 16·0 | 5⅜ | 835 | 8¾ | 5,862 | 14 | 38,416 | 20½ | 176,400 |
| 2⅛ | 20·4 | 5½ | 915 | 8⅞ | 6,204 | 14¼ | 41,250 | 20¾ | 185,400 |
| 2¼ | 25·6 | 5⅝ | 1,001 | 9 | 6,561 | 14½ | 44,226 | 21 | 194,480 |
| 2⅜ | 31·8 | 5¾ | 1,093 | 9⅛ | 6,922 | 14¾ | 47,350 | 21¼ | 203,800 |
| 2½ | 39·1 | 5⅞ | 1,191 | 9¼ | 7,321 | 15 | 50,625 | 21½ | 213,540 |
| 2⅝ | 47·5 | 6 | 1,296 | 9⅜ | 7,725 | 15¼ | 54,100 | 21¾ | 223,730 |
| 2¾ | 57·2 | 6⅛ | 1,407 | 9½ | 8,145 | 15½ | 57,750 | 22 | 234,256 |
| 2⅞ | 68·3 | 6¼ | 1,526 | 9⅝ | 8,582 | 15¾ | 61,520 | 22¼ | 245,025 |
| 3 | 81·0 | 6⅜ | 1,652 | 9¾ | 9,037 | 16 | 65,536 | 22½ | 256,036 |
| 3⅛ | 95·4 | 6½ | 1,785 | 9⅞ | 9,509 | 16¼ | 69,690 | 22¾ | 268,320 |
| 3¼ | 112 | 6⅝ | 1,926 | 10 | 10,000 | 16½ | 74,110 | 23 | 279,840 |
| 3⅜ | 130 | 6¾ | 2,076 | 10¼ | 11,025 | 16¾ | 78,720 | 23¼ | 292,680 |
| 3½ | 150 | 6⅞ | 2,234 | 10½ | 12,166 | 17 | 83,521 | 23½ | 304,700 |
| 3⅝ | 173 | 7 | 2,401 | 10¾ | 13,363 | 17¼ | 88,560 | 23¾ | 318,100 |
| 3¾ | 198 | 7⅛ | 2,577 | 11 | 14,641 | 17½ | 93,850 | 24 | 331,380 |
| 3⅞ | 225 | 7¼ | 2,763 | 11¼ | 16,027 | 17¾ | 99,225 | 24¼ | 345,740 |
| 4 | 256 | 7⅜ | 2,958 | 11½ | 17,503 | 18 | 104,970 | 24½ | 360,000 |
| 4⅛ | 290 | 7½ | 3,164 | 11¾ | 19,072 | 18¼ | 110,930 | 24¾ | 375,770 |
| 4¼ | 326 | 7⅝ | 3,380 |  |  |  |  |  |  |

## Table CXCVIII.—Tons of Water delivered per hour with a Loss of Head of 5 lbs. through Pipes of various Sizes and Lengths $\left(Q=10\sqrt{\frac{d^5\times 5}{L}}\right)$.

| Bore of Pipe. | Length of Pipe in feet. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 5 | 10 | 15 | 20 | 30 | 40 | 50 | 60 | 70 | 80 | 100 | 120 | 140 | 160 |
| ins. | | | | | | | | | | | | | | |
| 1 | 5·00 | 3·54 | 2·89 | 2·50 | 2·04 | 1·77 | 1·57 | .. | .. | .. | .. | .. | .. | .. |
| 1½ | 23·1 | 19·5 | 15·9 | 13·8 | 11·3 | 9·75 | 8·76 | .. | .. | .. | .. | .. | .. | .. |
| 2 | 56·5 | 40·0 | 32·8 | 28·4 | 21·4 | 20·1 | 18·1 | .. | .. | .. | .. | .. | .. | .. |
| 2½ | 98·8 | 70·0 | 57·1 | 50·0 | 40·3 | 35·0 | 31·2 | 28·5 | 26·4 | .. | .. | .. | .. | .. |
| 3 | 156 | 110 | 90·0 | 77·9 | 63·6 | 55·2 | 49·3 | 45·0 | 41·6 | .. | .. | .. | .. | .. |
| 3½ | 229 | 162 | 133 | 115 | 93·5 | 81·0 | 72·4 | 66·1 | 61·2 | 57·3 | 51·2 | .. | .. | .. |
| 4 | 320 | 227 | 185 | 160 | 131 | 113 | 101 | 92·4 | 85·5 | 80·0 | 71·6 | 65·3 | 60·5 | 56·6 |
| 4½ | 430 | 304 | 248 | 215 | 175 | 152 | 136 | 124 | 115 | 108 | 96·1 | 87·8 | 81·2 | 76·0 |
| 5 | 562 | 400 | 325 | 281 | 229 | 200 | 177 | 162 | 150 | 140 | 126 | 115 | 106 | 100 |
| 6 | 882 | 624 | 509 | 441 | 360 | 312 | 279 | 255 | 236 | 221 | 197 | 180 | 167 | 156 |
| 7 | 1,298 | 909 | 749 | 649 | 529 | 459 | 411 | 374 | 346 | 324 | 290 | 265 | 245 | 230 |
| 8 | 1,811 | 1,281 | 1,047 | 906 | 739 | 641 | 573 | 523 | 484 | 453 | 405 | 370 | 342 | 320 |
| 9 | 2,433 | 1,702 | 1,406 | 1194 | 993 | 861 | 770 | 702 | 650 | 609 | 544 | 497 | 460 | 430 |
| 10 | 3,119 | 2,237 | 1,827 | 1582 | 1290 | 1119 | 1000 | 912 | 845 | 791 | 707 | 646 | 600 | 559 |
| 12 | 4,989 | 3,530 | 2,883 | 2495 | 2036 | 1765 | 1578 | 1439 | 1332 | 1248 | 1116 | 1019 | 943 | 882 |
| 14 | 7,335 | 5,190 | 4,239 | 3670 | 3000 | 2600 | 2320 | 2116 | 1960 | 1834 | 1640 | 1500 | 1386 | 1300 |
| 16 | 10,241 | 7,247 | 5,919 | 5124 | 4180 | 3623 | 3253 | 2955 | 2733 | 2561 | 2290 | .. | .. | .. |
| 18 | 13,305 | 9,730 | 7,946 | 6880 | 5611 | 4865 | 4350 | 4000 | 3674 | .. | .. | .. | .. | .. |
| 20 | 17,900 | 12,660 | 10,336 | 9000 | 7300 | 6330 | 5658 | .. | .. | .. | .. | .. | .. | .. |

*N.B.*—The amount delivered by any other head $p_1$ can be found by multiplying the above by $\sqrt{0\cdot2\times p_1}$ or $0\cdot45\sqrt{p_1}$.

## Table CXCIX.—Hyperbolic Logarithms of Numbers up to 232.

| No. | Logthm. | No. | Logthm. | No. | Logthm. | No. | Logthm. |
|---|---|---|---|---|---|---|---|
| 1·10 | 0·0953 | 1·60 | 0·4700 | 2·20 | 0·7885 | 3·20 | 1·1632 |
| 1·15 | 0·1398 | 1·65 | 0·5008 | 2·30 | 0·8329 | 3·30 | 1·1939 |
| 1·20 | 0·1823 | 1·70 | 0·5306 | 2·40 | 0·8755 | 3·40 | 1·2238 |
| 1·25 | 0·2231 | 1·75 | 0·5596 | 2·50 | 0·9163 | 3·50 | 1·2528 |
| 1·30 | 0·2624 | 1·80 | 0·5878 | 2·60 | 0·9555 | 3·60 | 1·2809 |
| 1·35 | 0·3001 | 1·85 | 0·6152 | 2·70 | 0·9932 | 3·70 | 1·3083 |
| 1·40 | 0·3365 | 1·90 | 0·6419 | 2·80 | 1·0296 | 3·80 | 1·3350 |
| 1·45 | 0·3720 | 1·95 | 0·6678 | 2·90 | 1·0647 | 3·90 | 1·3610 |
| 1·50 | 0·4055 | 2·0 | 0·6931 | 3·00 | 1·0986 | 4·00 | 1·3863 |
| 1·55 | 0·4382 | 2·10 | 0·7419 | 3·10 | 1·1314 | 4·20 | 1·4351 |

## Table CXCIX.—Hyperbolic Logarithms—*continued.*

| No. | Logthm. | No. | Logthm. | No. | Logthm. | No. | Logthm. |
|---|---|---|---|---|---|---|---|
| 4·40 | 1·4816 | 24 | 3·1781 | 66 | 4·1898 | 109 | 4·6921 |
| 4·60 | 1·5261 | 25 | 3·2189 | 67 | 4·2050 | 110 | 4·7013 |
| 4·80 | 1·5686 | 26 | 3·2581 | 68 | 4·2197 | 111 | 4·7103 |
| 5·00 | 1·6094 | 27 | 3·2958 | 69 | 4·2343 | 112 | 4·7193 |
| 5·20 | 1·6487 | 28 | 3·3322 | 70 | 4·2487 | 113 | 4·7283 |
| 5·40 | 1·6864 | 29 | 3·3673 | 71 | 4·2630 | 114 | 4·7370 |
| 5·60 | 1·7228 | 30 | 3·4012 | 72 | 4·2768 | 115 | 4·7458 |
| 5·80 | 1·7519 | 31 | 3·4341 | 73 | 4·2907 | 116 | 4·7555 |
| 6·00 | 1·7918 | 32 | 3·4656 | 74 | 4·3043 | 117 | 4·7630 |
| 6·20 | 1·8245 | 33 | 3·4965 | 75 | 4·3178 | 118 | 4·7715 |
| 6·40 | 1·8563 | 34 | 3·5267 | 76 | 4·3310 | 119 | 4·7798 |
| 6·60 | 1·8871 | 35 | 3·5557 | 78 | 4·3568 | 120 | 4·7883 |
| 6·80 | 1·9094 | 36 | 3·5838 | 79 | 4·3696 | 121 | 4·7966 |
| 7·00 | 1·9459 | 37 | 3·6111 | 80 | 4·3823 | 122 | 4·8049 |
| 7·20 | 1·9741 | 38 | 3·6379 | 81 | 4·3947 | 123 | 4·8130 |
| 7·40 | 2·0015 | 39 | 3·6639 | 82 | 4·4069 | 124 | 4·8211 |
| 7·60 | 2·0281 | 40 | 3·6893 | 83 | 4·4191 | 125 | 4·8291 |
| 7·80 | 2·0541 | 41 | 3·7139 | 84 | 4·4311 | 126 | 4·8372 |
| 8·00 | 2·0794 | 42 | 3·7378 | 85 | 4·4429 | 127 | 4·8450 |
| 8·20 | 2·1041 | 43 | 3·7615 | 86 | 4·4546 | 128 | 4·8528 |
| 8·40 | 2·1282 | 44 | 3·7849 | 87 | 4·4661 | 129 | 4·8607 |
| 8·60 | 2·1518 | 45 | 3·8069 | 88 | 4·4781 | 130 | 4·8683 |
| 8·80 | 2·1748 | 46 | 3·8290 | 89 | 4·4894 | 131 | 4·8761 |
| 9·00 | 2·1972 | 47 | 3·8504 | 90 | 4·5005 | 132 | 4·8837 |
| 9·20 | 2·2192 | 48 | 3·8714 | 91 | 4·5115 | 133 | 4·8913 |
| 9·40 | 2·2407 | 49 | 3·8921 | 92 | 4·5225 | 134 | 4·8987 |
| 9·60 | 2·2618 | 50 | 3·9123 | 93 | 4·5334 | 135 | 4·9060 |
| 9·80 | 2·2824 | 51 | 3·9321 | 94 | 4·5440 | 136 | 4·9134 |
| 10·0 | 2·3026 | 52 | 3·9515 | 95 | 4·5546 | 137 | 4·9208 |
| 11·0 | 2·3979 | 53 | 3·9706 | 96 | 4·5652 | 138 | 4·9281 |
| 12·0 | 2·4849 | 54 | 3·9893 | 97 | 4·5756 | 139 | 4·9353 |
| 13·0 | 2·5649 | 55 | 4·0077 | 98 | 4·5857 | 140 | 4·9424 |
| 14·0 | 2·6391 | 56 | 4·0256 | 99 | 4·5958 | 141 | 4·9496 |
| 15 | 2·7081 | 57 | 4·0433 | 100 | 4·6052 | 142 | 4·9567 |
| 16 | 2·7726 | 58 | 4·0606 | 101 | 4·6159 | 143 | 4·9638 |
| 17 | 2·8332 | 59 | 4·0779 | 102 | 4·6258 | 144 | 4·9708 |
| 18 | 2·8904 | 60 | 4·0947 | 103 | 4·6354 | 145 | 4·9777 |
| 19 | 2·9444 | 61 | 4·1111 | 104 | 4·6451 | 146 | 4·9846 |
| 20 | 2·9957 | 62 | 4·1274 | 105 | 4·6548 | 147 | 4·9914 |
| 21 | 3·0445 | 63 | 4·1433 | 106 | 4·6642 | 148 | 4·9982 |
| 22 | 3·0911 | 64 | 4·1592 | 107 | 4·6737 | 149 | 5·0049 |
| 23 | 3·1353 | 65 | 4·1746 | 108 | 4·6829 | 150 | 5·0115 |

## Table CXCIX.—Hyperbolic Logarithms—*continued.*

| No. | Logthm. | No. | Logthm. | No. | Logthm. | No. | Logthm. |
|---|---|---|---|---|---|---|---|
| 151 | 5·0181 | 172 | 5·1483 | 193 | 5·2636 | 213 | 5·3622 |
| 152 | 5·0246 | 173 | 5·1541 | 194 | 5·2688 | 214 | 5·3669 |
| 153 | 5·0312 | 174 | 5·1598 | 195 | 5·2740 | 215 | 5·3716 |
| 154 | 5·0378 | 175 | 5·1656 | 196 | 5·2791 | 216 | 5·3763 |
| 155 | 5·0443 | 176 | 5·1714 | 197 | 5·2842 | 217 | 5·3809 |
| 156 | 5·0507 | 177 | 5·1771 | 198 | 5·2893 | 218 | 5·3855 |
| 157 | 5·0572 | 178 | 5·1827 | 199 | 5·2943 | 219 | 5·3900 |
| 158 | 5·0636 | 179 | 5·1883 | 200 | 5·2992 | 220 | 5·3945 |
| 159 | 5·0698 | 180 | 5·1939 | 201 | 5·3043 | 221 | 5·3991 |
| 160 | 5·0760 | 181 | 5·1994 | 202 | 5·3093 | 222 | 5·4037 |
| 161 | 5·0822 | 182 | 5·2050 | 203 | 5·3142 | 223 | 5·4081 |
| 162 | 5·0884 | 183 | 5·2104 | 204 | 5·3190 | 224 | 5·4125 |
| 163 | 5·0945 | 184 | 5·2158 | 205 | 5·3240 | 225 | 5·4171 |
| 164 | 5·1006 | 185 | 5·2212 | 206 | 5·3289 | 226 | 5·4216 |
| 165 | 5·1068 | 186 | 5·2266 | 207 | 5·3337 | 227 | 5·4259 |
| 166 | 5·1129 | 187 | 5·2320 | 208 | 5·3385 | 228 | 5·4302 |
| 167 | 5·1189 | 188 | 5·2374 | 209 | 5·3433 | 229 | 5·4346 |
| 168 | 5·1248 | 189 | 5·2427 | 210 | 5·3480 | 230 | 5·4390 |
| 169 | 5·1307 | 190 | 5·2480 | 211 | 5·3522 | 231 | 5·4434 |
| 170 | 5·1366 | 191 | 5·2532 | 212 | 5·3574 | 232 | 5·4477 |
| 171 | 5·1425 | 192 | 5·2584 |  |  |  |  |

## Table CXCIXa.—Power Transmitted per Revolution through Standard Shafts by Rule diameter $= \sqrt[3]{\frac{\text{S.H.P.}}{\text{R}}} \times 64.$

| Diameter of Shaft. | S.H.P. ÷ R. | Diameter of Shaft. | S.H.P. ÷ R. | Diameter of Shaft. | S.H.P. ÷ R. | Diameter of Shaft. | S.H.P. ÷ R. |
|---|---|---|---|---|---|---|---|
| ins. |  | ins. |  | ins. |  | ins. |  |
| 5·00 | 1·953 | 8·0 | 8·000 | 12·0 | 27·00 | 18·0 | 91·25 |
| 5·25 | 2·262 | 8·25 | 8·766 | 12·5 | 30·52 | 18·5 | 98·92 |
| 5·50 | 2·600 | 8·50 | 9·594 | 13·0 | 34·32 | 19·0 | 107·2 |
| 5·75 | 2·970 | 8·75 | 10·47 | 13·5 | 38·44 | 19·5 | 115·8 |
| 6·00 | 3·375 | 9·00 | 11·41 | 14·0 | 43·25 | 20·0 | 125·0 |
| 6·25 | 3·814 | 9·25 | 12·36 | 14·5 | 47·62 | 20·5 | 134·6 |
| 6·50 | 4·291 | 9·50 | 13·39 | 15·0 | 52·74 | 21·0 | 144·7 |
| 6·75 | 4·805 | 9·75 | 14·49 | 15·5 | 58·19 | 21·5 | 155·2 |
| 7·0 | 5·363 | 10·0 | 15·62 | 16·0 | 64·00 | 22·0 | 166·4 |
| 7·25 | 5·954 | 10·5 | 18·08 | 16·5 | 70·19 | 22·5 | 178·0 |
| 7·5 | 6·591 | 11·0 | 20·80 | 17·0 | 76·76 | 23·0 | 190·1 |
| 7·75 | 7·274 | 11·5 | 23·77 | 17·5 | 83·74 | 24·0 | 216·0 |

# NOMENCLATURE AND DEFINITIONS AS ESTABLISHED BY THE BRITISH MARINE ENGINEERING DESIGN AND CONSTRUCTION COMMITTEE.

## (A) Boilers: Definitions of Various Sorts.

*A Main Boiler* is one whose special and general function is to supply steam to the engines engaged in driving the propellers of the ship, or to the auxiliaries necessary for the proper working of the same, and whose normal supply of feed water is from a surface condenser.

*An Auxiliary Boiler* is one used generally for purposes other than that of supplying the main propelling machinery, and, while it may be so used on an emergency, it is not a necessary part of the main boiler installation as used when at full service speed. It is, however, supplied with feed water from a surface condenser, or with other water equally pure, except on emergency.

*Winch (Donkey) Boilers* are those having no connection with the main boilers but employed on winches, cranes, and other appliances generally outside the engine- and boiler-rooms, and whose normal supply of feed water is from the sea or other impure water which will deposit solid matter in the boiler.

*Vertical Boilers* are those whose cylindrical shells are placed with the axis vertical or nearly so, and have the fire grate in the furnace at or near to the bottom.

## (B) Tubes and Pipes: Definitions of Terms used.

A straight hollow cylindrical body made of copper, brass, iron, steel, or other malleable metal shall be called a *Tube*.

A tube when bent, flanged or otherwise worked and treated mechanically, or made up for service shall be called a *Pipe*.

The thickness of all tubes and pipes over 18 L.S.G. thick shall be expressed in 100*ths of an inch*.

## (C) Shafting: Nomenclature and Definitions of.

1. *Line of Shafting.*—A line of shafting in a screw steamship consists of the motor shaft at one end and the shaft to which the propeller is attached at the other, together with the shafts intermediate between them.

2. *Motor Shaft.* — The motor shaft in the case of reciprocating engines is the crank shaft, and in turbine engines is the rotor shaft. With an electrical drive it is the rotor or armature shaft.

3. *Tail Shaft.*—The shaft to which the propeller is fitted shall be called the tail shaft.

4. *Tube Shaft.*—When the tail shaft is entirely outboard, the shaft which passes into the inside of the ship shall be called the tube shaft.

5. *Outboard Intermediate Shaft.*—When there is a shaft interposed between the tail shaft and the tube shaft, it shall be called the outboard intermediate shaft.

6. *Intermediate Shafts.*—The shafts connecting the tube shaft with the motor shaft shall be called intermediate shafts.

7. *Thrust Shaft.*—The shaft having collars formed upon it for the purpose of resisting the longitudinal thrust on the shaft shall be called the thrust shaft.

8. *Couplings.*—The flanges forged solid with a shaft, or welded to the body as with iron and scrap steel shafts, and the other portions forged with, or those which are fitted to the shafts and are removable, whereby the shaft may be connected to other shafts, shall be called couplings.

9. *Coupling Bolts.*—When two shafts are connected together by fitted bolts with heads or with taper bodies, secured in place by nuts, they shall be called coupling bolts.

*Driver Bolts.*—Those bolts having plain cylindrical bodies without heads, which are secured to the coupling of one shaft by nuts, and are free to move longitudinally in the holes of the coupling of the adjacent shaft, shall be called driver bolts.

10. With geared turbines the shafts having on them the pinions and gear wheels, shall be called as follows:—

*Motor Spindle.*—Motor spindle or motor shaft is the axle or shaft of the rotor of a turbine or electric motor.

*First Pinion Spindle.*—The first pinion spindle is that coupled to the motor spindle and having on it the pinion which gears into the wheel of the next spindle or shaft.

*Second Pinion Spindle.*—When there is double gearing the second pinion spindle is that which has on it the wheel driven by the first pinion and also the second pinion which drives the wheel on the next spindle or shaft.

*Main Wheel Shaft.*—The main wheel shaft is that which carries the final wheel and is coupled to the intermediate shafting.

When there are more than one set of gearing provided to each line of shafting on account of there being more than one turbine to each, these spindles will have added to their designations high pressure, medium, or low pressure (H.P., M.P., L.P.).

11. *Crank Webs.*—In the case of cranks shafts, the arms shall be called crank webs.

Table CC.—Horse-power Transmissible by Solid Steel Shafts from Turbines, Electric, or other Motors having Uniform Torque (B.M.E.D. & C. Committee Rule, $d^3 = S.H.P. \times 64 \div$ Revs. p.m.).

| Diameter of Shaft. | Revolutions per Minute. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 75. | 100. | 125. | 150. | 175. | 200. | 225. | 250. | 275. | 300. | 325. | 350. | 375. | 400. |
| ins. | | | | | | | | | | | | | | |
| 5·0 | 146 | 195 | 244 | 293 | 342 | 391 | 439 | 488 | 537 | 586 | 635 | 683 | 732 | 781 |
| 5·5 | 195 | 260 | 325 | 390 | 455 | 520 | 585 | 650 | 715 | 780 | 845 | 910 | 975 | 1,040 |
| 6·0 | 254 | 339 | 424 | 509 | 594 | 679 | 764 | 849 | 933 | 1,018 | 1,103 | 1,188 | 1,273 | 1,358 |
| 6·5 | 322 | 429 | 536 | 644 | 751 | 858 | 966 | 1,073 | 1,180 | 1,287 | 1,395 | 1,502 | 1,610 | 1,717 |
| 7·0 | 402 | 536 | 670 | 804 | 938 | 1,072 | 1,206 | 1,340 | 1,474 | 1,608 | 1,742 | 1,876 | 2,010 | 2,144 |
| 7·5 | 495 | 660 | 825 | 990 | 1,155 | 1,320 | 1,485 | 1,650 | 1,815 | 1,980 | 2,145 | 2,310 | 2,475 | 2,640 |
| 8·0 | 600 | 800 | 1,000 | 1,200 | 1,400 | 1,600 | 1,800 | 2,000 | 2,200 | 2,400 | 2,600 | 2,800 | 3,000 | 3,200 |
| 8·5 | 720 | 960 | 1,200 | 1,440 | 1,680 | 1,920 | 2,160 | 2,400 | 2,640 | 2,880 | 3,120 | 3,360 | 3,600 | 3,840 |
| 9·0 | 854 | 1,140 | 1,425 | 1,709 | 1,994 | 2,279 | 2,564 | 2,848 | 3,134 | 3,418 | 3,702 | 3,988 | 4,272 | 4,556 |
| 9·5 | 1,000 | 1,334 | 1,667 | 2,001 | 2,334 | 2,668 | 3,002 | 3,335 | 3,668 | 4,002 | 4,335 | 4,669 | 5,002 | 5,336 |
| 10·0 | 1,170 | 1,560 | 1,950 | 2,340 | 2,730 | 3,120 | 3,510 | 3,900 | 4,290 | 4,680 | 5,070 | 5,460 | 5,850 | 6,240 |
| 10·5 | 1,356 | 1,808 | 2,260 | 2,712 | 3,164 | 3,616 | 4,068 | 4,520 | 4,972 | 5,424 | 5,876 | 6,328 | 6,780 | 7,232 |
| 11·0 | 1,560 | 2,080 | 2,600 | 3,120 | 3,640 | 4,160 | 4,680 | 5,200 | 5,720 | 6,240 | 6,760 | 7,280 | 7,800 | 8,320 |
| 11·5 | 1,785 | 2,380 | 2,975 | 3,570 | 4,165 | 4,760 | 5,355 | 5,950 | 6,545 | 7,130 | 7,725 | 8,320 | 8,915 | 9,510 |
| 12·0 | 2,025 | 2,700 | 3,375 | 4,050 | 4,725 | 5,400 | 6,075 | 6,750 | 7,425 | 8,100 | 8,775 | 9,450 | 10,125 | 10,800 |
| 13 | 2,574 | 3,432 | 4,290 | 5,148 | 6,006 | 6,864 | 7,722 | 8,580 | 9,438 | 10,296 | 11,154 | 12,012 | 12,870 | 13,728 |
| 14 | 3,216 | 4,288 | 5,360 | 6,432 | 7,504 | 8,576 | 9,648 | 10,720 | 11,792 | 12,864 | 13,936 | 15,008 | 16,080 | 17,152 |
| 15 | 3,954 | 5,272 | 6,590 | 7,908 | 9,226 | 10,544 | 11,862 | 13,180 | 14,498 | 15,816 | 17,134 | 18,452 | 19,770 | 21,088 |
| 16 | 4,800 | 6,400 | 8,000 | 9,600 | 11,200 | 12,800 | 14,400 | 16,000 | 17,600 | 19,200 | 20,800 | 22,400 | 24,000 | 25,600 |
| 17 | 5,757 | 7,676 | 9,595 | 11,514 | 13,433 | 15,352 | 17,271 | 19,190 | 21,109 | 23,028 | 24,947 | 26,866 | 28,785 | 30,700 |
| 18 | 6,834 | 9,112 | 11,390 | 13,668 | 15,946 | 18,224 | 20,502 | 22,780 | 25,058 | 27,336 | 29,614 | 31,892 | 34,170 | 36,448 |
| 19 | 8,040 | 10,720 | 13,400 | 16,080 | 18,760 | 21,440 | 24,120 | 26,800 | 29,480 | 32,160 | 34,840 | 37,520 | 40,200 | 42,880 |
| 20 | 9,375 | 12,500 | 15,625 | 18,750 | 21,875 | 25,000 | 28,125 | 31,250 | 34,375 | 37,500 | 40,625 | 43,750 | 46,875 | 50,000 |

*N.B.*—If from a two-crank 90° engine, 70 per cent. should be the limit. From a three-crank 120° or a balanced four-crank, 78 per cent.; and from a four-crank 90°, 75 per cent. For smooth water service, 110 per cent. each.

# APPENDIX A.

## Rules of the British Corporation for the Survey and Registry of Shipping (Machinery).

## GENERAL CONDITIONS OF CLASSIFICATION.

### (Section 31.)

The construction of the machinery for Steam Vessels, and of donkey boilers for Sailing Vessels, which are intended for classification with the British Corporation, is to be carried out under the supervision and to the satisfaction of the Surveyors, and before the work is proceeded with, detailed plans and particulars of the boilers and superheaters, copies of the specifications of boiler material, and full information regarding the engines, together with plans showing proposed arrangement of engine seating and holding-down bolts, plans of bilge and ballast pumping arrangements, evaporators, feed-water filters and heaters, and particulars of sizes of steam and feed pipes, are to be submitted for the approval of the Committee. The clear space between bottom of boilers and top of tank of floors to be stated in all cases, and is not to be less than 18 inches, without the special sanction of the Committee. Upon satisfactory completion in accordance with the Rules, the machinery will be entered in the Register Book "M.B.S.*" (Machinery British Standard—Special Survey).

## BOILERS. (Section 32.)

1. The Rules following are intended to apply to the construction of steel boilers; where boilers are to be made of iron, they will be specially considered by the Committee.

2. The quality of steel to be used in the construction of boilers must be of the best "mild" quality, made on the "Open Hearth" system, free from hammer-dressing, cracks, and surface defects. It must have a workmanlike finish, and be capable of standing the following tests:—

3. All samples shall be selected by the Surveyor, properly marked, and tested in his presence prior to despatch, and the tensile strength and ductility shall be determined from test pieces cut lengthwise or crosswise from the rolled material, in accordance with the requirements specified in Sec. 3, par. 3:—

The tensile strength and ductility shall be determined from test pieces cut lengthwise or crosswise from the finished material, and when material is annealed or otherwise treated before despatch the test pieces shall be similarly and simultaneously treated before testing. Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test piece, and any straightening of test pieces which may be required shall be done cold. The parallel part of the tensile test piece shall not be less than 9 inches in length, and the width shall not exceed 1½ inches for material more than $\frac{35}{40}$ inch in thickness, from $\frac{35}{40}$ to $\frac{15}{40}$ it shall not exceed 2 inches, and under $\frac{15}{40}$ the width shall not be more than 2½ inches. For round bars the parallel part may be either nine or four times the diameter of the test piece, but the sectional area must not be less than ¼ square inch; the bars may be tested the full size as rolled. Should a tensile test piece break outside the middle half of its gauge length the test may, at the maker's option, be discarded and another test made of the same plate or bar. Test pieces for bends shall not be less than 1½ inches wide, and for small bars the whole section may be used. For temper bend tests the samples are to be heated uniformly to a blood red and quenched in water at a temperature not exceeding 80° Fahrenheit. The colour shall be judged indoors in the shade. The rough edge or arris caused by shearing may be removed from all cold bends and from temper bends above $\frac{19}{40}$ inch in thickness by filing or grinding, pieces 1 inch thick and above may be machined, but the test pieces shall receive no other preparation. The pieces are not to be annealed unless the material from which they are cut is similarly and simultaneously treated.

4. One tensile test and one temper or cold bend test shall be taken from each plate as rolled. For plates exceeding 2½ tons in weight one tensile and one bend test shall be taken from each end—one bend test to be temper and the other cold. For angle bars, rivet bars, and stay bars at least two tensile tests shall be taken from each charge; but when the number of the bars, as rolled, from one charge exceeds 15, an additional tensile test shall be made for each batch of 15 bars, or portion thereof, with the exception of round bars 1¾ inch diameter and under, when the number shall be 50 in place of 15. A temper or cold bend test shall be made from each angle bar as rolled, a temper and cold bend test shall be made from every fifteen stay bars as rolled, from each charge, but for rivet bars no bend tests will be required.

5 The tensile breaking strength of plates for shells shall be between the limits of 28 and 32 tons per square inch if not otherwise specified. For plates intended for flanging or welding and for combustion chambers and furnaces, also for bars for combustion chamber stays

and rivet bars, the tensile breaking strength shall be between the limits of 26 and 30 tons per square inch. In the case of material for purposes in which tensile strength is not important, the tensile test may be dispensed with, and the bend test only made if so specified and approved.

All material which has satisfactorily passed the requirements must be clearly stamped by the manufacturer in at least two places on each finished bar or plate, thus—B.C., signifying that the material has satisfactorily complied with the tests above described. Material the tensile breaking strength of which is between the limits of 26 and 30 tons, and complies with the corresponding requirements for elongation, should be marked $\frac{\text{B.C.}}{\text{F.}}$ to signify that it is of "cold flanging" quality, and material which has been approved on bend tests only is to be marked $\frac{\text{B.C.}}{\text{B.}}$ so that it may be readily identified. No material bearing these brands is to be forwarded from the steel works until the prescribed tests have been made by the Surveyors and the mill sheets signed by them. The material must also be legibly stamped with the manufacturer's name or trade mark, and the place where made.

6. The elongation on a gauge length of 8 inches or 8 diameters shall not be less than 20 per cent. for material required to have a tensile breaking strength of 28 to 32 tons per square inch, and not less than 23 per cent. for material required to have a tensile breaking strength of 26 to 30 tons per square inch. For material under $\frac{15}{40}$ in thickness the respective elongations may be 3 per cent. less. Where short test pieces are used for round bars, the elongations on $3\frac{1}{2}$ diameters shall be 25 and 29 per cent. respectively. Rivet bars shall have an elongation of not less than 25 and 31 per cent. on gauge lengths of 8 and $3\frac{1}{2}$ diameters respectively, intermediate proportions of length to diameter to have intermediate percentages.

7. Cold and temper bend test pieces must withstand, without fracture, being doubled over until the internal radius is equal to $1\frac{1}{2}$ times the thickness of the sample and the sides are parallel. Bending tests may be made by pressure or by blows.

8. The stamping of the material, rejection for defects, the issuing of advice notes, &c., must be carried out in the same way as for ship steel, the Rules for which are as follows:—

In the event of any test piece failing to meet the requirements, unless two further tests prove satisfactory, the plate or bar represented by such sample piece must be rejected. Further tests of pieces taken from the same charge shall then be made, and in case of these also failing, the whole of the material produced from such charge is to be rejected.

All material which has satisfactorily passed the requirements must be clearly stamped by the manufacturer, as described in the paragraph beginning at line 6 on this page.

Before the mill sheets are signed, the manufacturer must furnish the Surveyor with a certificate guaranteeing that the material has been made by the "Open Hearth" process, and that it has withstood satisfactorily the tests above described. The following form of certificate will be accepted if printed on each mill sheet with the name of the firm, and initialed by the test house manager:—

> "We hereby certify that the material described below has been made by the 'Open Hearth' process, and is that which has been satisfactorily tested in the presence of the Surveyors in accordance with the tests of the British Corporation Registry."

When it is found necessary from failure in testing, surface defects, or other sufficient cause, to reject any of the material submitted for testing, the Surveyor is to satisfy himself that the stamp of the Corporation upon the rejected material is obliterated by centre punch marks, thus—B·C.

The maker must adopt a system of marking the ingots, billets, slabs, test pieces, and finished material, which will enable all material to be traced to its tests and original charge. The Surveyor must be given every facility for tracing the material to the respective charges from which it is made, and he should be furnished with two copies of advice notes of such material as may have been satisfactorily dealt with, for his signature—one to be forwarded by the manufacturer to the boiler-maker, the other to be retained for the use of the Surveyor in attendance at the boiler works.

Where steel is not produced in the works at which it is rolled, a certificate shall be supplied to the Surveyor deputed to witness the testing of the material, stating the Open Hearth process by which it is made, the name of the steel maker who supplied it, and the numbers of the charges, for reference to the books of the steel maker. The number of the charge shall be marked on each plate or bar for the purpose of identification.

The foregoing tests shall be made at the place of manufacture prior to the despatch of the material; but in the event of any of the material proving unsatisfactory in the process of being worked, it shall be rejected, and such further tests of material from the same charge may be made as the Surveyor in attendance may consider desirable, and material then found defective shall also be rejected, notwithstanding any previous certificate of satisfactory testing.

9. In addition to the tests for rivet steel specified above, the manufactured rivets are to be tested, hot and cold, by bending, crushing under the steam hammer, or in such other way as may be required, a sufficient number of sample rivets being taken indiscriminately for the purpose. The shank must stand being doubled together cold, without fracture, and the rivet head must stand being flattened hot until its diameter is $2\frac{1}{2}$ times the diameter of the shank. The rivets are not to be used until such testing has been carried out to the satisfaction of the Surveyors.

10. All plates which have been welded, or locally heated for furnace work, and plates in which the rivet holes are punched instead of being drilled, must be annealed after this work is carried out. Steel stay bars which have been worked in the fire must be subsequently annealed, and in no case are steel stays to be welded.

11. **Strength Calculations.**—The sizes and arrangement of the different parts for a given working pressure, or the working pressure suitable to a given size and arrangement of material, must be in accordance with the following formulæ and rules. All manhole or similar openings in boilers are to be efficiently compensated:—

12. **Cylindrical Shells of Steel** whose tensile is S tons:—

$$W = \frac{(T-2) \times E \times S}{D \times C}.$$

Where C = 2·75 when the longitudinal seams are fitted with double butt straps of equal width, and of thickness at least equal to that obtained from the formula for butt straps (see par. 13).
C = 2·85 when the double butt straps are of unequal width, *i.e.* one strap not covering the outer row of rivets, with the thickness as before.
C = 2·95 when the longitudinal seams are lap joints.
T = thickness of shell plate, in 32nds of an inch.
E = the least percentage of strength of longitudinal joints, found as follows:—

For the plate at the joint, $E = \frac{p-d}{p} \times 100.$

For the rivets at the joint, $E = \frac{n \times a}{p \times t} \times \frac{r}{s} \times 100.$

Where $p$ = pitch of rivets in outer rows, in inches.
$d$ = diameter of rivet holes, in inches.
$n$ = number of rivets used per pitch.
$a$ = sectional area of one rivet, in square inches.
$t$ = thickness of plate, in inches.
$r$ = 23 for steel and 19 for iron rivets.
$s$ = minimum tensile strength of shell plates, as specified.

Where rivets are in double shear, $a \times 1·875$ is to be used instead of $a$.

D = greatest internal diameter of shell, in inches.
W = working pressure, in lbs. per square inch.

13. **Double Butt Straps.**—The thickness of each plate is to be in accordance with the following formula:—

$$T_1 = T\frac{5(p-d)}{8(p-kd)}.$$

Where $T_1$ = thickness of outer strap, in 32nds of an inch.
$T_1 + 4$ = thickness of inner strap.
$T$ = thickness of shell plate, in 32nds of an inch.
$p$ = pitch of rivets in outer rows, in inches.
$d$ = diameter of rivet holes, in inches.
$k$ = ratio of pitch of rivets in outer rows to that in centre rows.

14. **Flat Surfaces** supported by Stays :—

(*a*) $\text{W.P.} = \dfrac{C \times (T-1)^2}{P^2 + p^2}$ ; and thickness $T = \sqrt{\dfrac{\text{W.P.}(P^2+p^2)}{C}} + 1.$

(*b*) Flat surfaces not exposed to flame, fitted with doublings between the rows of supporting stays, the doublings being at least 0·6 of the thickness of the plates to which they are riveted :—

$$\text{W.P.} = \frac{(T-1)^2 + 0{\cdot}55\,t^2}{P^2+p^2} \times C\text{; and } T = \sqrt{\frac{\text{W.P.}(P^2+p^2)}{C} - 0{\cdot}55\,t^2} + 1.$$

(*c*) Flat surfaces not exposed to flame, strengthened by outside doublings or washers on the stays, and fitted with nuts outside and inside :—

$$\text{W.P.} = \frac{(T-1)^2 + (C\times t)^2 \times 100}{P^2+p^2}\text{; and } T = \sqrt{\frac{\text{W.P.}(P^2+p^2)}{100} - (C\times t)^2} + 1.$$

Where W.P. = working pressure, in lbs. per square inch.
$T$ = thickness of plate, in 32nds of an inch.
$t$ = thickness of doublings or washers, in 32nds of an inch.
$P$ = distance between the rows of stays, in inches.
$p$ = pitch of stays in the rows, in inches.
$C = 50$ for plates fitted with screwed stays having riveted heads.
$C = 75$ for plates fitted with screwed stays having nuts outside.
$C = 100$ for plates fitted with stays having nuts inside and outside.
$C = 0{\cdot}15$ for loose washers not less than 0·6 the thickness of plate and three times the diameter of stay.
$C = 0{\cdot}35$ when the washers are riveted to plates and in diameter two-thirds pitch.
$C = 0{\cdot}55$ when there are doubling strips riveted to plates.
$C = 0{\cdot}85$ when there is a doubling plate over the whole surface.

*Note.*—For front plates in the steam space, which are not protected against the direct action of the flame, the constants given above are to be reduced 20 per cent. Stays which are screwed into plates of less thickness than $\frac{7}{16}$ inch are not to have riveted heads, but are to be fitted with nuts.

### 15. Tube Plates with Tubes in Nests :—

$$\text{W.P.}=\frac{80\times(\text{T}-1)^2}{\text{P}^2+p^2}\text{; and T}=\sqrt{\frac{\text{W.P.}(\text{P}^2+p^2)}{80}}+1.$$

T = Thickness of plate in 32nds of an inch.

$$\text{W.P.}=\frac{875\times(\text{P}-d)\times\text{T}}{\text{L}\times\text{P}}\text{; and T}=\frac{\text{W.P.}\times\text{L}\times\text{P}}{(\text{P}-d)\times875}.$$

P = greatest pitch of stay tubes from centre to centre, in inches.
$p$ = least pitch of stay tubes from centre to centre, in inches.

For wide spaces between nests of tubes see par. 14.

When girders are fitted to the tops of combustion chambers, the thickness of the tube plates is not to be less than given by the preceding formula or than is found from the formula :—

$$\frac{\text{W}\times\text{L}\times\text{P}}{(\text{P}-d)\times1800}=\text{T}.$$

Where W = working pressure in lbs. per square inch.
L = width of combustion chamber over the plates, in inches.
P = horizontal pitch of tubes, in inches.
$d$ = inside diameter of plain tube, in inches.
T = thickness of tube plate, in sixteenths of an inch.

### 16. Stays :—

$$\sqrt{\frac{\text{S}\times\text{W}}{\text{C}}}+\tfrac{1}{8}=\text{D}.$$

Where S = surface, in square inches, supported by the stay.
W = working pressure in lbs. per square inch.
D = effective diameter of stay, in inches.
C = 8250 for steel screwed stays, and for iron screwed and longitudinal stays made from tested iron bars. Steel stays are to comply with the tests specified above. Iron stays are to be of the best quality and are to have a tensile breaking strength of not less than 20 tons per square inch. Iron bars for stays which require to be tested are to have a tensile strength of not less than 21½ tons per square inch; the elongation on a gauge length of 8 diameters is not to be less than 20 per cent. nor 27 per cent. on a gauge length of 3½ diameters.
C = 6500 for untested iron screwed stays.
C = 9500 for longitudinal steel stays.
C = 7000 for longitudinal iron stays.
C = 5000 for welded iron stays.

17. **Stay Tubes** are not to be subjected to a greater stress than 7500 lbs. per square inch.

18. **Circular Furnaces.**—Thickness of plain furnaces, combustion chamber bottoms, and of furnaces with Adamson rings pitched more than 20 inches apart:—

$$\frac{51{\cdot}5}{D}\left(18{\cdot}75\,T-(L\times 1{\cdot}03)\right)=W.$$

Suspension bulb furnaces to be:—

$$W.P.=\frac{510\times(T-1)}{D};\ \text{and}\ T=\frac{W.P.\times D}{510}+1.$$

Corrugated, ribbed, and suspension furnaces (Fox, Purves, Morison, Deighton, and Brown's cambered):—

$$W.P.=\frac{480(T-1)}{D};\ \text{and}\ T=\frac{W.P.\times D}{480}+1.$$

Where T = thickness of plate, in 32nds inch.
L = length of furnace, or combustion chamber bottom between the points of support, in inches.
D = smallest outside diameter of furnace, in inches.
W.P. = working pressure, in lbs. per square inch.

19. **Girders for Combustion Chamber Tops:**—

$$\frac{C\times d^2\times T}{(L-P)\times D\times L}\times\frac{(n+1)^2}{n(n+2)}=W\ \text{when}\ n=2,\ 4,\ \text{or}\ 6.$$

$$\frac{C\times d^2\times T}{(L-P)\times D\times L}=W\ \text{when}\ n=1,\ 3,\ \text{or}\ 5.$$

Where C = 495 for steel and 450 for iron.
$d$ = depth of girder at the centre, in inches.
T = thickness of girder, in inches.
D = distance from centre to centre of girders, in inches.
L = length from tube plate to tube plate, or from tube plate to back of combustion chamber, in inches.
$n$ = number of stays fitted with each girder.
P = pitch of stays supported by girder, in inches.

20. **General Construction.**—Each boiler must have at least one glass water gauge, two test cocks direct on the boiler shell, or an additional glass water gauge, and one steam pressure gauge. Where the pillar and connections of the water gauge are 2 inches or more in diameter, the test cocks may be fitted thereon. Double-ended boilers to have these fittings at each end. One salinometer cock to each boiler. Where a pillar is attached by pipes to steam and water spaces, cocks should be fitted to the boiler at the ends of these pipes. When water gauge pillars and their boiler connections are less than 2 inches diameter and no cocks on the boiler connections, the passage in the pillar between the gauge cocks is to be closed so as to permit of each end being blown through separately.

21. A stop-valve is to be fitted on each boiler, so that any one of a series of boilers may be worked independently. The neck of the stop-valve to be as short as possible. All boiler and engine stop-valves to be tested to at least twice the W.P.

22. Two safety-valves required for each main boiler, and one for each superheater, and they must be tested under steam and set to a pressure not more than 3 per cent. in excess of the intended W.P. The combined area of the valves is to be sufficient to prevent the steam accumulating to more than 10 per cent. of the W.P., during fifteen minutes full firing with main engines stopped. If the boilers be supplied with forced draught, the valve area must be such that the same conditions may be met.

23. Easing gear is to be so arranged that the safety-valves on any one boiler may be lifted without interfering with those on any other boiler.

24. All boiler blow-off pipes are to have cocks or valves on the hull plating fitted with spigots extending through the plating, and plate flanges round same on the outside; in addition, blow-off cocks or valves are to be fitted to the boiler. (See also sec. 33, par. 12.)

25. Manhole doors in boilers to be not less than 16 ins. by 12 ins.

26. Headers for water tube boilers and superheaters are to be tested by a hydraulic pressure of three times, and other parts by one of double W.P. The tubes to be of solid drawn steel and stand being flattened cold until the sides are not further apart than twice the thickness of the tube; they must also withstand expansion cold without cracking or splitting, until the diameter is increased by 10 per cent. in tubes $\frac{3}{16}$ thick or less, and 8 per cent. in tubes of greater thickness. The finished tubes should not exceed 27 tons per sq. in. tensile strength and should show an elongation of 20 per cent. on a length of 8 inches; at least one tube in fifty is to be subjected to flattening and expansion tests and all the tubes to a hydraulic pressure of 1500 lbs. per sq. in. The thicknesses of the tubes in inches are not to be less than given by the formula:—

$$T = \frac{D \times W}{6000} + \frac{1}{16}$$

where D = internal diameter of tube in inches,
and W = working pressure in lbs. per square inch.
T = thickness of tube in inches.

The two rows of tubes which are next the fire in water tube boilers are to be increased at least 10 per cent. in thickness.

**Tubes** for water tube boilers and superheaters are to be of solid drawn steel. (See present par. 26.)

27. Upon completion, the boilers are to be tested by hydraulic pressure to 1·5 the intended working pressure + 50 lbs., and after being placed in position in the vessel, they must be efficiently secured by brackets and stays to prevent any fore and aft or athwartship movement. Superheaters to be tested to twice the working pressure.

It is strongly recommended, because of the rapid corrosion which takes place in material near them, that the boilers be kept as high as possible above the floors or tank top, and that the under side of the boilers be efficiently insulated. (See sec. 31.)

28. Donkey boilers need not have more than one safety-valve, provided the valve area be not less than half a square inch for each foot of grate surface. In other respects the requirements for donkey boilers are the same as for main boilers.

## ·ENGINES. (Sec. 33.)

1. **Shafting.**—The minimum diameters of crank, thrust, propeller, and intermediate shafts may be found from the following formula, except where the ratio of length of stroke to distance between main bearings is unusual, when they will receive special consideration:—

$$D = \sqrt[3]{\frac{P \times L^2 \times S}{B}} \times C.$$

Where D = diameter of shaft in inches.
P = absolute pressure, *i e.* boiler pressure + 15 lbs.
S = stroke of engine, in inches.
L = diameter of low-pressure cylinder, in inches.
B = value as given in Table B.
C = 1·0 for crank and thrust shafts.
C = 0·95 for intermediate shafts.
C = for propeller shafts to be taken from the following Table:—

### Table C.

| Co-efft. of Displ. of vessel at $\frac{4}{5}$ moulded depth. | Ratio of Diameter of Propeller to Diameter of Crankshaft. | | | | | |
|---|---|---|---|---|---|---|
| | 13 | 14 | 15 | 16 | 17 | 18 |
| ·6 | 1·0 | 1·01 | 1·02 | 1·03 | 1·04 | 1·05 |
| ·62 | 1·01 | 1·02 | 1·03 | 1·04 | 1·05 | 1·06 |
| ·64 | 1·02 | 1·03 | 1·04 | 1·05 | 1·06 | 1·07 |
| ·66 | 1·03 | 1·04 | 1·05 | 1·06 | 1·07 | 1·08 |
| ·68 | 1·04 | 1·05 | 1·06 | 1·07 | 1·08 | 1·09 |
| ·70 | 1·05 | 1·06 | 1·07 | 1·08 | 1·09 | 1·10 |
| ·72 | 1·06 | 1·07 | 1·08 | 1·09 | 1·10 | 1·11 |
| ·74 | 1·07 | 1·08 | 1·09 | 1·10 | 1·11 | 1·12 |
| ·76 | 1·08 | 1·09 | 1·10 | 1·11 | 1·12 | 1·13 |
| ·78 | 1·09 | 1·10 | 1·11 | 1·12 | 1·13 | 1·14 |
| ·80 | 1·10 | 1·11 | 1·12 | 1·13 | 1·14 | 1·15 |

The value of the divisor C in the formula depends on the ratio $\frac{L^2}{H^2}$ where L = diameter of low-pressure cylinder and H of high-pressure cylinder, in inches:—

## Table B.

| $\frac{L^2}{H^2}$ | Two Cranks at 90° Compound or Quadruple, also Three Cranks at 120° Quadruple Expansion. | Three Cranks at 120°, Triple Expansion. | Four Cranks at 90° Quadruple Expansion. |
|---|---|---|---|
| Ratio 3 | 9,910 | .. | .. |
| ,, 3⅛ | 10,160 | .. | .. |
| ,, 3¼ | 10,410 | .. | .. |
| ,, 3⅜ | 10,660 | .. | .. |
| ,, 3½ | 10,910 | .. | .. |
| ,, 3⅝ | 11,160 | .. | .. |
| ,, 3¾ | 11,410 | .. | .. |
| ,, 3⅞ | 11,660 | .. | .. |
| ,, 4 | 11,910 | .. | .. |
| ,, 4⅛ | 12,160 | .. | .. |
| ,, 4¼ | 12,410 | .. | .. |
| ,, 4⅜ | 12,660 | .. | .. |
| ,, 4½ | 12,910 | 13,650 | .. |
| ,, 4¾ | 13,375 | 14,160 | .. |
| ,, 5 | 13,840 | 14,670 | .. |
| ,, 5¼ | 14,305 | 15,180 | .. |
| ,, 5½ | 14,770 | 15,690 | .. |
| ,, 5¾ | 15,235 | 16,200 | .. |
| ,, 6 | 15,700 | 16,710 | .. |
| ,, 6½ | 16,630 | 17,730 | .. |
| ,, 7 | 17,560 | 18,630 | .. |
| ,, 7½ | 18,410 | 19,530 | .. |
| ,, 8 | 19,260 | 20,430 | 22,660 |
| ,, 8½ | 20,110 | 21,330 | 23,660 |
| ,, 9 | 20,960 | 22,200 | 24,660 |
| ,, 9½ | 21,750 | 23,070 | 25,660 |
| ,, 10 | 22,540 | 23,940 | 26,580 |
| ,, 10½ | 23,330 | 24,810 | 27,500 |
| ,, 11 | 24,120 | 25,660 | 28,420 |
| ,, 11½ | 24,900 | 26,500 | 29,340 |
| ,, 12 | 25,680 | 27,340 | 30,260 |

Intermediate ratios to have intermediate divisors.

The least diameter of shafts for direct-coupled and geared turbine engines may be obtained from the following formula:—

$$D = \sqrt[3]{\frac{P \times 65}{R}} \times C.$$

Where D = diameter of intermediate shafts.
P = shaft horse power.
C = 1 R = revolutions per minute.
C = 1·05 D for intermediate thrust shafts between collars
C = 1·1 D propeller shafts.

With geared turbines where the revolutions per minute of the propeller are comparatively low, the value of C for the propeller shaft is not to be less than 1·1 D, nor than the values of C given in Table C, substituting the diameter of intermediate shafting for diameter of crank shaft in obtaining the ratio of propeller to shaft.

Thrust and propeller shafts may be gradually tapered off to the diameter of the intermediate shafting. The central hole through hollow shafting may have a diameter equal to one-third that of the intermediate shafting without increasing the diameters of the shafting required by the formula.

2. The webs of built crank shafts are to be keyed as well as shrunk on to the shaft, and the diameter of the shaft should be increased in way of the web to make up the loss of sectional area at the key way; efficient dowel pins should be fitted in the crank pins.

3. Propeller shafts must be forged from selected scrap iron, rolled iron bars, or a single steel ingot. It is recommended that shaft liners be fitted in one length, that the inboard ends of all shaft liners be tapered, and that the space between the after end and the propeller boss be made watertight.

4. **Forgings.**—Shafts, piston rods, connecting rods, and other important working parts of the machinery, are to be made from selected scrap iron or ingot steel in accordance with the following requirements. Other material, such as nickel steel, etc., will be accepted after compliance with such special tests as may be imposed. All shafts, etc., are to be subjected to examination during the progress of manufacture, when rough turned and when finished.

5. Iron forgings are to be in tensile strength between the limits of 20 and 24 tons per square inch, and to show an elongation of 20 per cent. on a gauge length corresponding to 3½ diameters of the test piece. Samples, 1 inch square, are to stand being bent cold to 90° over a radius of 1½ inches without fracture.

6. Steel forgings for machinery are to be forged from the lower two-thirds of ordinary ingots made by the "open hearth" process; the material is to be tested, and the sectional area of the body of the forging, when it leaves the hammer, should not exceed one-fifth, and no part of the forging should be more than three-fourths the area of the original ingot. Forgings are to be properly annealed in an annealing furnace, and must be free from defect. The tensile strength of the forgings is not to exceed 35 tons per square inch, without special sanction, the elongations are to vary from 30 per cent. at 27 tons to 22 per cent. with 35 tons tensile strength, measured on test pieces of which the gauge length is not less than 3½ times their diameter; samples, 1 inch by ¾ inch, with edges rounded to $\frac{1}{16}$th radius, must stand being bent cold to 180° without fracture over a radius of ¼ inch for material under 32 tons tensile strength, and ⅜ inch between 32 and 35 tons tensile strength. The test pieces are to be cut lengthwise from a part of the forging which is not of less sectional area than the body of the forging, and they are not to be detached until they have been stamped by the Surveyors, and until the forging

has been annealed, in cases where subjected to the annealing process. One tensile and one bend test are to be taken from each forging, except in the case of forgings over 3 tons weight and of all propeller shafts, where tests will be required from each end of the forging. Where a number of small forgings are made from one ingot, tests representing each end of the ingot will be sufficient. All ingot steel forgings are to be legibly marked in such a way that they can be traced to the charge from which the material has been made, and these which have satisfactorily passed the requirements are to be clearly stamped "B.C.," and with the identification marks furnished by the Surveyor.

**7. Castings.**—Steel castings are to be made in accordance with the requirements of sec. 4, pars. 15–17, when their use is sanctioned for parts of the machinery which are usually forged. Where steel castings are used instead of the iron castings in ordinary use, they are to be subjected to percussive and hammering tests only. Castings for high-pressure cylinders of reciprocating engines are to be tested to 1·5 W.

Castings for cylinders of turbine engines are to be tested under hydraulic pressure after having been rough bored, to the following requirements:—

H.P. cylinders, admission end, are to be tested to 1·33 W
H.P. ,, exhaust end, ,, ,, ,, ,, W
I.P. ,, ,, ,, ,, ,, 1·5 V
L.P. ,, admission end, ,, ,, ,, ,, 1·5 V
L.P. ,, exhaust end, ,, ,, ,, ,, 30 lbs. per sq. inch.
Where W = working pressure of boilers.
V = pressure to which cylinder safety valves are to be adjusted.

The provisions in section 3 are as follows:—

(15) Where steel castings are intended to be used instead of forgings, every care is to be taken to avoid abrupt changes in sectional area, and each casting must be thoroughly annealed and subjected to percussive and hammering tests, as well as to tests for tensile strength and ductility. The castings should be made by the "open hearth" process, and are to be accurately moulded and free from defects. The material is to have a tensile strength of not less than 26 tons and not exceeding 35 tons per sq. in., with a corresponding elongation varying from 20 per cent. with 26 tons to 15 per cent. with 35 tons, measured on test pieces of which the length between the gauge points is not to be less than 3½ times the diameter, and the sectional area not less than ¼ sq. inch.

(16) The sample pieces for testing must have formed part of the actual casting submitted for approval, must have been subjected to similar and simultaneous annealing, and are not to be detached from the casting until after the annealing is completed, nor until they have been stamped by the Surveyor. The piece for ductility test should not be less than 1 inch by ¾ inch, with edges rounded to a radius of $\frac{1}{16}$ inch, and must stand being bent cold through an angle of 120°, over a radius of 1 inch, without any appearance of fracture.

(17) Castings of propeller frames, posts, single-plate rudder frames, brackets, and quadrants, must be raised to a height of 6 to 10 feet, according to the character and form of the castings, and dropped bodily on to a hard surface, holes being prepared to receive bosses or similar projections. When large propeller frames are cast in one piece, they may be dropped through an angle of 45° instead of being lifted bodily. Castings of a complex design may, at the discretion of the Surveyor, be exempted from the drop test, and where the material is thin the limits of elongation may be reduced by 5 per cent. All castings are to be slung clear of the ground and well hammered all over with a heavy sledge hammer to test the soundness of the material.

8. **General Construction.**—When the engine bed plate or thrust block is fastened direct to the tank, the top plating in way of same must not be less than $\frac{38}{40}$ inch thick and must be increased according to the size and power of the engines, and the double bottom efficiently stiffened under the thrust block, to the approval of the Committee. The holes for the holding-down bolts are to be tapped through the plate, and the bolts properly screwed in, and fitted with lock nuts underneath. The holding-down bolts must be kept as close to the angle bar as possible, and if they pass through the bar the flanges of the bars must be of sufficient breadth to take the nuts.

9. Two bilge and two feed pumps are to be fitted on the main engines in vessels of 180 feet length and upwards, and are to be so arranged that one of either set may be examined while the other is at work, or equivalent independent pumping arrangements are to be provided. In vessels less than 180 feet one bilge and one feed pump or equivalent independent pumps are to be provided; each bilge pump must be adapted to draw from any compartment of the vessel. All feed pumps worked by the main engines are to be fitted with spring-loaded relief valves. A bilge suction to the circulating pump, having a diameter about twice that of the engine room bilge suction, or a bilge injection valve, is to be fitted in each case.

10. The design and arrangement of pumps, valve chests, suction and delivery pipes, and all cocks and sea connections should be such as to prevent the possibility of water being run into the vessel accidentally.

11. All evaporators and high-pressure feed heaters must be satisfactorily tested by hydraulic pressure before leaving the works of the manufacturer. Feed-water filters between pumps and boilers to be tested to 20 per cent. above double working pressure.

12. All sea-cocks and valves connecting suction pipes are to be placed, if practicable, above the level of the stokehold and engine room platforms, and so as to be easily worked from those platforms. Sea-cocks must be fitted direct on to the hull plating and secured with tap bolts or with bolts having countersunk heads. Discharge pipes should be placed above the deep load line, with valves fitted in an accessible position on the vessel's side.

13. Donkey pumps adapted to draw from the sea and hotwell, to

supply the boilers with water through separate auxiliary feed pipes and check valves, to deliver water on deck, and to draw the bilge from each compartment, are to be fitted in all cases. When there are more main boilers than one, and the donkey pump is arranged to draw from the bottom of the boilers, for circulating purposes, the suction pipes are to have non-return valves on the boiler connections, and a shut-off valve or cock on the main pipe from the boilers to the pump.

14. All pipes extended through bunkers, or other compartments beyond the machinery space, must be well protected with strong casings.

15. **Steam and Feed Pipes** are to be of copper, wrought iron, or steel of thicknesses in accordance with the following formulæ. All main steam pipes must have efficient provision for expansion, and are to be so arranged that water cannot lodge in any part of them, or, if this be impracticable, so that they can be properly drained. Auxiliary steam pipes which exceed 4 inches in diameter, and all main and refrigerator machine steam pipes, are to be tested by hydraulic pressure, in the presence of the Surveyor, to at least twice the working pressure when made of copper, and to three times the working pressure when made of iron or steel; and all boiler feed pipes are to be subjected to a hydraulic pressure at least 20 per cent. greater than is required for steam pipes. When pipes have screwed flanges they are to be screwed with vanishing threads, and the thread is not to extend beyond the back of the flange collar.

$\frac{D \times W}{6000} + \frac{1}{16} =$ thickness in inches of brazed copper pipes.

$\frac{D \times W}{6000} + \frac{1}{32} =$ thickness in inches of solid drawn copper pipes.

$\frac{D \times W}{10{,}000} + \frac{1}{8} =$ thickness in inches of iron and steel pipes; but the minimum thickness for pipes which have to be bent, or are to have screwed ends, is to be $\frac{3}{16}$ inch.

$\frac{D \times W}{12{,}000} + \frac{1}{8} =$ thickness in inches of solid drawn steel pipes.

Where D = internal diameter of pipe inches.
W = working pressure for steam pipes.
W = boiler pressure × 1·2 for feed pipes.

16. **Spare gear**, consisting of the following articles, must be supplied to all vessels built to the requirements of the British Corporation:—

2 connecting rod top-end bolts.
2 ,, bottom-end bolts.
2 main-bearing bolts.
1 set of coupling bolts.
1 ,, feed and bilge pump valves.
1 ,, piston springs, if common springs are used.
1 safety-valve spring of each size fitted, but not fewer than 1 spare spring for each 6 safety-valves fitted.
½ set of fire bars for each boiler.

A number of bolts, nuts, and studs of assorted sizes, including at least six cylinder cover bolts or studs, and six valve chest cover bolts or studs.

Bar and plate iron in various sizes.

17. **Completion of Machinery.**—When all the connections are made on board, the safety-valves set, and the machinery completely fitted up, a trial under steam is to be made with the Surveyor in attendance, and if everything is then found to be working satisfactorily the Committee will grant a Certificate, and the machinery will be entered in the Register Book, under the Class of the Hull, thus—M.B.S.*, with date of completion.

## STEAM PUMPING ARRANGEMENTS. (Sec. 34.)

1. **Bilge Suctions in Vessels with Ordinary Floors.**—Not less than three steam pump suctions, of the sizes given in the Table, are to be fitted at the after end of each compartment, one being placed at the centre line and one in each wing. At the ends of the vessel, or in compartments where the rise of floor is considerable, one centre line suction will be accepted.

2. **Bilge Suctions in Vessels with Double Bottoms.**—There must be one suction in each wing at the after end of each compartment, of the size given in the Table for wing suctions with two suctions in each hold. Where wells are adopted, the suctions are to be as required for vessels with ordinary floors; but it is recommended that the wing suctions be placed outside the wells, and the openings into the wells fitted with non-return valves.

3. **Double Bottoms.**—At least one centre suction, of the size required by the Table for the size of the tank, is to be fitted at the after end of each compartment in the double bottom. In vessels where the rise of floor is less than 1 inch per foot, and the depth to the top of the midship erections is 75 per cent. or more of the vessel's breadth, centre and wing suctions should be fitted in the midship tanks.

4. **Peaks, and Deep Tanks.**—Where peaks or cargo spaces are used as ballast tanks, suctions of the size required by the Table for the size of tank are to be fitted. Cargo spaces used as ballast tanks are to have satisfactory arrangements for cutting off the bilge suctions when used for water and the ballast suctions when used for cargo; also for draining the tops of the tanks.

5. **Tunnel Wells** must in all cases be fitted with suctions, of not less size than is required for wing suctions with three suctions in each hold.

6. **The Main and Donkey Pumps** are to draw from all compartments, and in addition the donkey is to have a separate bilge suction in the engine room. The pipes connecting the bilge pumps with the valve chests are not to be of less diameter than required for centre suctions, and the ballast pump connections of not less diameter than required for a single suction to the largest tank suction in the vessel. The pumps must be of sufficient capacity to give a speed of water

through the pipes of not less than 400 feet per minute, under ordinary working.

7. **Bilge and Ballast Suction Pipes** are to be efficiently secured, and straps are to be fitted at the middle of the length of each range of pipes to prevent fore and aft movement. Efficient expansion joints are to be fitted, and where the connections at the ends of each range of pipes are made with lead bends, the radii of the bends and the distance between the centres of the radii should each be equal to three diameters, and the length of the bend to eight diameters of the pipe.

8. **All Roses and Boxes** to be placed so as to be easily accessible for examination and cleaning. It is recommended that each engine-room bilge suction should have a mud box placed above the level of the floor plates, with a tail pipe led from it into the bilges, and without the usual rose box in the bilges. Vessels under 180 feet in length should have provision made on the donkey pump for the attachment of a flexible hose, which should be supplied in addition to the engine-room requirements.

9. **Sounding Pipes** are to be fitted to each compartment and ballast tank, with a small doubling plate bedded in the cement under each pipe, for the rod to strike on. These pipes must be fitted, without bends, directly into the compartment intended to be sounded, and should in all holes extend to the upper deck.

10. **Air Pipes**, not less than 2 inches in diameter, are to be fitted at each corner of each ballast tank ; this requirement may be modified in the case of small tanks and increased for large tanks. Efficient arrangements must be made to permit of the air getting freely to the pipes while the tanks are being filled.

### Inside Diameters of Pipes (Minimum Requirements).

| BILGE SUCTIONS. | | | |
|---|---|---|---|
| LENGTH OF VESSEL. | Centre and all Engine Room Suctions. | Wing Suctions with Two Suctions in hold. | Wing Suctions with Three Suctions in hold. |
| | Inches. | Inches. | Inches. |
| Under 180 feet, . . . | 2 | ... | ... |
| 180 feet and under 230 feet, . | 2¼ | 2 | 2 |
| 230 ,, ,, 280 ,, . | 2½ | 2¼ | 2 |
| 280 ,, ,, 330 ,, . | 2¾ | 2½ | 2¼ |
| 330 ,, ,, 390 ,, . | 3 | 2¾ | 2½ |
| 390 ,, ,, 450 ,, . | 3¼ | 3 | 2½ |
| 450 ,, ,, 520 ,, . | 3½ | 3¼ | 2¾ |

Inside Diameters of Pipes (Minimum Requirements)—*contd.*

| TANK SUCTIONS. | | | |
|---|---|---|---|
| CAPACITY OF TANK. | Diameter of Suction. | CAPACITY OF TANK. | Diameter of Suction. |
| | Inches. | | Inches. |
| Under 20 tons, . . | 2½ | 270 tons and under 365 tons, | 5 |
| 20 tons and under 40 tons, | 2¾ | 365 ,, ,, 480 ,, | 5½ |
| 40 ,, ,, 60 ,, | 3 | 480 ,, ,, 625 ,, | 6 |
| 60 ,, ,, 85 ,, | 3¼ | 625 ,, ,, 800 ,, | 6½ |
| 85 ,, ,, 120 ,, | 3½ | 800 ,, ,, 1000 ,, | 7 |
| 120 ,, ,, 190 ,, | 4 | 1000 ,, ,, 1300 ,, | 7½ |
| 190 ,, ,, 270 ,, | 4½ | | |

## ELECTRIC LIGHTING. (Sec. 35.)

While the Committee do not consider it advisable at the present time to formulate definite rules for fitting vessels with electric-light installation, they are prepared to approve of such installations as may be carried out on well-considered and safe systems. It is suggested, therefore, that a complete specification of the apparatus and method of wiring proposed should be submitted for the consideration of the Committee, together with samples of the wires, switches, cut-outs, etc., if required.

In drawing up the specification, the following points should be kept in view:—

1. Dynamos, motors, main and branch cables should be so placed that the compasses will not be injuriously affected by the electric current. Tests should be made while the compasses are being adjusted, in order to prove that this condition has been satisfactorily fulfilled.

2. Conductors, switches, cut-outs, and hull connections, if any, should be so arranged as to be always and easily accessible.

3. Switches and cut-outs should be on non-inflammable bases—preferably of porcelain—and the switches should be of the quick-break pattern.

4. Cut-outs should be fitted in every case at each reduction in size of wire in the single wire system, and on both wires where double wiring is adopted. It is recommended that tin wire cut-outs of standard sizes be used, the number of sizes being kept as small as possible.

5. Hull connections for "single wire" systems should be made with brass screws of large surface, carefully fitted.

6. Conductors should be made of the best copper, and the insulation resistance should not be less than 600 megohms per mile—a certificate from the manufacturer to this effect to be furnished if desired. Vulcanized rubber insulation is strongly recommended, but other materials may be adopted if possessing equal insulation resistance and durability.

No single wire should be smaller in diameter than No. 18, or larger than No. 16 standard wire gauge; where sizes above No. 16 are required the conductors should be formed of small wires stranded together.

The current density in the conductors should not exceed 1000 amperes per square inch.

All conductors should be protected by efficient casings, or lead covered, or armoured, and the covers on wood casings should be portable and fastened with screws. Conductors which are exposed to excessive heat or damp, or which are led through bunkers or cargo spaces, should be lead-sheathed, and in engine room, stokehold, and tunnel they should be further protected by armour.

Special care should be taken to protect conductors from damp and chafe where they pass through deck or bulkheads.

7. Joints in main cables should be avoided wherever possible, and, when absolutely necessary, they should be so made that the watertightness and insulation resistance of the cables is not affected by them.

Ordinary joints in branches should be properly soldered and thoroughly insulated, resin only being used as the flux for soldering. It is strongly recommended that such a system of distributing and auxiliary switch boards be adopted as will render all ordinary joints unnecessary.

8. A voltmeter should be fitted in every case, and an ammeter where there is more than one dynamo.

9. A final test should be made when the installation is complete, when the insulation resistance over the whole or any part of the system should not be below the following :—

| | | | |
|---|---|---|---|
| Installations of 25 lights, | . . . | 500,000 ohms. |
| ,, 50 ,, | . . . | 250,000 ,, |
| ,, 100 ,, | . . . | 125,000 ,, |
| ,, 500 ,, | . . . | 25,000 ,, |
| ,, 1000 ,, | . . . | 12,500 ,, |

For intermediate numbers of lights the insulation resistance should be correspondingly proportionate.

For alternating currents the minimum insulation resistance should be twice that given in the Table.

## PERIODICAL SURVEY OF ENGINES AND BOILERS.

The engines and boilers of all vessels classed with the Corporation will be required to undergo Special Periodical Surveys at the same times as the Special Surveys on the hull. In cases of accident involving considerable repair and an extensive examination of the machinery, such examination may, with the sanction of the Committee, be considered equivalent to a Special Survey.

At each Special Survey the cylinders or turbine, pistons, valves, pumps, evaporators, thrust block and condenser, main and tunnel bearings, shafting, and the steam-steering gear must be opened up for examination, and such other parts of the machinery as may be considered necessary are to be examined.

The arrangements for pumping from the several holds, as well as from the engine and boiler space, are to be inspected, and while the vessel is in dry dock, all openings to the sea, together with the cocks and valves in connection with the same, examined; in addition, all iron and steel fastenings of sea-cocks and valves to the shell plating should be removed for examination at each Special Survey No. 3. The propeller shaft should be drawn at least once every two years, and more frequently if considered necessary by the Surveyor, but when liners are fitted solid in one length the shaft need only be drawn once every three years; the Committee are, however, prepared to consider representation from owners as to special circumstances which may modify these requirements in particular cases.

When the after bearing is worn down $\frac{1}{4}$ inch with shafts not exceeding 9 inches in diameter, $\frac{5}{16}$ when over 9 and not exceeding 12 inches, and $\frac{3}{8}$ with shafts over 12 inches in diameter, the bearing must be rebushed.

At each Special Survey, the boilers, superheaters, and safety-valves are to be carefully examined inside and outside, and the safety-valves set to the working pressure, and the main steam pipes should be tested by hydraulic pressure every four years in case of brazed copper pipes, and every six years in case of solid drawn copper pipes and of iron and steel pipes; copper pipes should be annealed before being tested. If at any of these surveys the Surveyor considers it desirable, the actual thickness of plates and strength of stays are to be ascertained in order to determine the future working pressure, and the boilers and superheaters tested by hydraulic pressure.

In addition to the requirements of the Special Periodical Surveys, when six years old, and annually thereafter, the boilers, superheaters, and safety-valves must be carefully examined and the valves set to the working pressure.

If it be found desirable, upon inspection, that any part of the engines or boilers should be examined again within a short period, it will be necessary for the owner to comply with the Committee's requirements in this respect.

The donkey boilers of Sailing Vessels are to be subject to Special Periodical and Annual Survey in accordance with the foregoing requirements.

# APPENDIX B.

## BUREAU VERITAS RULES

### For Machinery.*

**Classification.**—Mechanically propelled vessels will only be granted a class after the machinery has been examined and certified by an Engineer Surveyor of the Bureau Veritas as being in accordance with the rules.

The Special Survey mark ✠ will only be granted in the Register when the boilers and machinery have been constructed under the special inspection of an Engineer Surveyor of the Bureau Veritas.

The above provisions apply also to sailing vessels fitted with auxiliary machinery.

To obtain Special Survey of machinery and boilers the following provisions must be complied with :—

1. Make application in writing to District Surveyor.

2. Submit plans of boilers and safety-valves, and give full particulars of materials to be used.

3. Give Surveyors every opportunity and facility for making full inspections of all parts during manufacture and fixing.

Hand to Surveyors duplicates of all orders for material, and see that they have facilities for testing all material.

4. Notify Surveyors where hydraulic tests as described below can be witnessed :—

Cylinders of Compound, Triple, and Quadruple engines,—H.P. cylinders to be tested to boiler pressure plus 85 lbs. per sq. in. ; L.P. cylinders to 43 lbs. per sq. in. ; I.P. cylinders of triples to 0·8 boiler pressure ; first intermediate cylinders of quadruples to boiler pressure ; and second intermediate cylinders of quadruples to half boiler pressure.

H.P. turbine casings to be tested to one and a quarter times boiler pressure ; L.P. turbine casings to a quarter boiler pressure at admission ends and 28·5 lbs. per sq. in. at exhaust ends ; astern turbine casings to one and a quarter boiler pressure H.P. and 43 lbs. L.P. end.

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* These rules are summarised, and not given word for word.

5. Condensers to be tested to 28·5 lbs. per sq. in.

6. Boilers to be tested to twice working pressure up to 142 lbs. per sq. in., but for all higher working pressures the test pressure is to be working pressure plus 142 lbs. per sq. in.

7. Steam and feed pipes to be tested to twice working pressure, and other steam pipes to twice maximum pressure to which they will be subject. These tests may be made after fitting on board.

9. Boilers, after being fitted on board, to be tested again to once and a half working pressure up to 85 lbs. per sq. in., but for all higher working pressures to working pressure plus 85 lbs. per sq. in.

11. The Surveyor will afterwards attend a steam trial and adjust the safety-valves, and will also proceed with the vessel on her trial trip and satisfy himself as to the efficient working of the whole of the machinery.

**Maintenance of Class: Annual and Periodical Surveys.**—Periodical Surveys of machinery must take place every four years for vessels of the first division, and every three years for vessels of the second and third divisions.

Boilers over twelve years old must undergo the periodical Survey with the corresponding tests every two years. In countries where the testing of boilers is regulated by law, the Surveyors are authorised to conform to the legal requirements.

Propellers must be removed and propeller shafts drawn for examination at least every two years.

Surveyor to be notified whenever boiler, propeller, or tail shaft is removed; also all repairs to be carried out to satisfaction of Surveyor and reported by him to the Administration.

A Survey is to be held in all cases of accident to machinery, and the master of the vessel must notify Surveyor; failing such notification, class may be withdrawn.

An annual Survey will be mainly external so far as main and auxiliary engines are concerned, though the Surveyor is empowered to open up any part that he may think necessary. As regards boilers, the annual Survey will be both external and internal, and the Surveyor will see boilers under steam and readjust safety-valves if necessary.

At periodical Surveys the Surveyor will require to have cylinders opened and pistons removed, and slide-valves and main bearings opened up; and he will also carefully examine condensers, pumps, pipes, cocks and valves, propeller and other shafting and propeller. He will also make a complete examination and hydraulic test of boilers, and may order repairs, lowering of working pressure, or may even condemn boilers. The hydraulic test to be to one and a half times working pressure, as described above.

When boilers twelve years old are being examined the Surveyor will probably require to drill various parts to ascertain thicknesses, and he may require the partial removal of lagging from boilers and steam pipes.

On completion of a periodical Survey, the Surveyor will attend a steam trial and satisfy himself as to the efficiency of the whole of the machinery.

**Construction and Arrangement of Engines.**—All machinery must be so securely fixed that the motions of the vessel will not cause anything to come adrift; sufficient space must be provided around and above engines and in tunnels, and sufficient ventilation given to enable the engineers to give proper attention whilst engines are running, and also to effect necessary examinations and repairs. Guard plates and handrails must be fitted where required.

Turning gear must, and steam reversing gear should, be fitted to all engines over 500 horse-power.

Condensers are to be of strong construction and fitted with suitable manholes and handholes for examination and cleaning.

Valve-chests, cocks and pipe connections must be so arranged (except ash cocks and water service cocks) that sea water cannot be run into the ship.

All discharge valves to be fitted direct on vessel's skin plating above load line if possible, in accessible positions, and should be capable of being closed in harbour.

All pumping pipes should be easily accessible, and none should be carried through the bunkers without proper protection.

Bilge pumps and donkey engines to pump from all compartments except fore and after peaks; and if there be no well, in holds with double bottoms, a suction must be fitted in each wing gutter. In addition to the suctions from the common valve boxes the engine bilge pumps must be fitted with a direct suction from the engine-room bilges.

When the circulating pump is fitted with a bilge suction, care must be taken to arrange so that sea-water cannot run into the ship; and suitable gratings must be fitted at inlets.

Steam pipes must be so fitted as to allow of expansion; drain cocks must be fitted where necessary; and steam pipes on deck must be protected by casings.

Satisfactory means of communication between engine room and bridge must be fitted.

In vessels under the French flag all pipes must be painted with colours indicating their purposes or uses.

**Construction and Arrangement of Boilers.**—Boilers must be so secured that neither the rolling of the vessel nor the shock of collision will move them: they must be arranged in spaces of sufficient size for easy working and repair, and the said spaces must be properly ventilated, must have two easily accessible exits, and must have the overhead openings fitted with gratings and shutters or covers.

When donkey boilers and auxiliary machinery are placed outside the main engine and boiler rooms, the spaces provided must be enclosed by steel bulkheads and the decks beneath must be plated and caulked; the spaces must also be properly ventilated.

In all steam ships provision must be made for preventing coal or ashes getting down through stokehole flooring; and in those of more than 800 horse-power a steam ash-hoisting apparatus must be fitted.

Coal bunkers must be kept at least one foot away from boilers or chimney unless insulating material is fitted; and proper ventilating arrangements must be fitted, especially in vessels with more than one deck, Steam pipes must not pass through bunkers, and if water pipes or electric leads do pass through they must be protected by strong casings.

Oil fuel may be carried in the double-bottom, bunkers, peaks, or other suitable compartments if various conditions, laid down in detail in the Rules, are complied with. If the oil fuel has a lower flash-point than 150° F., the proposed arrangements must be specially submitted to the Administration for approval.

The water-level should, ordinarily, be at least six inches above the tops of the combustion chambers, but in small boilers this may be reduced to four inches.

The minimum level must be distinctly marked as near as possible to the water-gauges.

At least two spring safety-valves of approved design and of area given by following formula must be fitted direct to each main boiler; and they must be provided with easing gear workable both from stokehole and deck, and also with drain-cocks.

When A is area of valve in square inches, that must be provided per square foot of grate surface, and P is the boiler pressure in lbs. per square inch,

$$A = \frac{8 \cdot 7}{\sqrt[4]{(P - 18)^3}}.$$

Where forced draught is fitted, the areas of safety-valves must be increased in proportion to the increased evaporative power of the boilers.

For boilers with grate surface not exceeding five square feet only one valve will be required, but no valve of less than 1¼ inches diameter will be accepted. All single-seated safety-valves must be capable of lifting at least one-quarter of the diameter of the valve.

All steam receptacles receiving from the boilers of more than 3·5 cubic feet capacity must have a safety-valve when the effective pressure is over 4·27 lbs.

Each boiler (except donkey boilers, etc.) shall be supplied by two separate feed pumps (one at least independent of main engines), pipes, and check-valves, each of sufficient size to feed the boiler in all circumstances.

Every single-ended boiler must be fitted with two water-level indicators, one at least being a glass water-gauge; the second may consist of a set of test cocks (ordinarily three in number, but for small boilers two). Double-ended boilers must have a water-gauge at each end. When test cocks are fitted for pressures over 114 lbs. per square inch, long rods or handles for working them must be supplied.

A stop valve or cock must be fitted direct to the boiler shells at every outlet for steam, and the main stop valves should, where possible, be arranged to be worked from deck.

Every cylindrical boiler should be fitted with both a surface and a bottom blow-off valve or cock attached direct to the shell; but for water-tube boilers a bottom blow-off only will be required. Blow-off cocks with spigots and brass or galvanised iron outside washers must also be fitted on the skin plating of the vessel. Arrangements should be made to ensure that the opening and closing of blow-off cocks can be properly regulated; and where one pipe connects several boilers, cocks or valves for preventing the passage of water from one boiler to another must be fitted.

Every single-ended boiler must be fitted with a pressure-gauge marked to show working pressure, and every double-ended boiler must be fitted with two such gauges. Gauges to be placed in view of firemen, to have cocks for shutting them off from the boilers, and to have lamps fitted near them.

Each boiler must be furnished with a fitting terminating in a flange 1¼ inch diameter and ¼ inch thick for attaching a standard gauge, and also with a suitable connection for making the hydraulic test.

Every boiler must be provided with manholes, mudholes, and sight holes, conveniently arranged for inspection and cleaning. No manhole to be less than 12 inches by 16 inches, and all are to be fitted with compensating rings.

Cast-iron must not be used for manhole doors.

## BOILER SHELLS AND STAYS.

### *Circular Shells and Steam-holders with Internal Pressure.*

A riveted joint may fail through the tearing of the plate or butt-strap between the rivets, the shearing of all the rivets, or by a combination of the two. The following formulæ apply to these several cases. The plate thickness and the diameter of rivets to be applied to have the highest values which each formula would give separately.

### I. *Rupture through Plate.*

The formulæ for working pressure and plate thicknesses are in this case:—

$$\left.\begin{aligned} P &= \frac{2\,a\,R\,(t-0{\cdot}04)}{D} \\ \text{and} \quad t &= \frac{P\,D}{2\,a\,R} + 0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad . \quad (\text{I.})$$

Where P = allowed working pressure, above atmosphere, in pounds per square inch.

D = greatest inside diameter of boiler shell, or steam-holder in inches.

$t$ = thickness of shell plates in inches. $t - 0{\cdot}04$ inch represents the thickness left after a reduction of 0·04 inch through corrosion.

R = the tensile stress, in pounds per square inch, which will be allowed in the plate. The value of R will be the breaking strength divided by 4, the latter figure representing the factor of safety for the plate after it has been corroded away by 0·04 inch.

If the actual breaking strength happens to be known by tests carried out to the Administration's satisfaction, it may be applied for finding R; but when, as usual, it is not known, the value of R will be:—

**For Steel:** the 4th part of the lower limit of tensile strength chosen by the designer, which in such case is to be stated when a boiler design is submitted for approval.

The table annexed shows the values of 2 R for various tensile strengths.

$a$ = ratio of the resistance of the plate left between the holes, to that of the full plates. It will be determined from the following expression:—

$$a=\frac{p-d}{p}.$$

Where $p$ = pitch of rivets in outer row, in inches (see figs. 1 and 2).
$d$ = diameter of rivet holes, in inches.

TABLE SHOWING THE VALUES OF 2 R. = 1120 × TENSILE.

*In Formulæ* (I.) *and* (IV.) *for various Tensile Strengths of the Material.*

| Tensile Strength of Plates in tons per sq. inch. | Value of 2 R. | Tensile Strength of Plates in tons per sq. inch. | Value of 2 R. |
|---|---|---|---|
| 32 | 35,800 | 25·5 | 28,600 |
| 31·5 | 35,300 | 25·0 | 28,000 |
| 31 | 34,700 | 24·5 | 27,400 |
| 30·5 | 34,200 | 24·0 | 26,900 |
| 30 | 33,600 | 23·5 | 26,300 |
| 29·5 | 33,000 | 23·0 | 25,800 |
| 29 | 32,500 | 22·5 | 25,200 |
| 28·5 | 31,900 | 22·0 | 24,600 |
| 28 | 31,400 | 21·5 | 24,100 |
| 27·5 | 30,800 | 21·0 | 23,500 |
| 27 | 30,200 | 20·5 | 22,900 |
| 26·5 | 29,700 | | |
| 26 | 29,100 | | |

*N.B.*—Boiler designs submitted for approval of Administration must state lowest tensile of steel for various parts and also shearing strength of rivets; if these are not stated, minimum tensile of all plates (including furnace) and stays will be taken as 26 tons, and steel rivets will be understood to comply with Administration's requirements.

In every case copy of specification of material, with name of makers, must be supplied to Surveyor through whom designs are submitted for approval.

### II. *Rupture through Rivets.*

In this case the following are the formulæ for finding the allowed working pressure, or required rivet section :—

$$\left.\begin{aligned} P&=\frac{2\,A\,S}{D\,l} \\ \text{and} \qquad A&=\frac{P\,D\,l}{2\,S} \end{aligned}\right\} \quad \ldots\ldots\ldots \quad \text{(II.)}$$

Where P and D have the same meaning as before, and

$l$ = the length, in inches, of the identical parts into which a riveted joint can be subdivided. In most cases $l$ is the pitch of the rivets in the outer rows (figs. 1 and 2).

In general it depends upon the system of joint adopted.

S = the maximum shearing stress, in pounds per square inch, which will be allowed on the rivets. It will be the 4th part of the actual shearing resistance of the material, which should always be ascertained by tests, when possible. If the actual shearing resistance of the rivet bars is not known, it will be assumed to amount to 0·8 of their tensile strength, and the value of $s$ will be one-fifth part of the lower tensile limit adopted by the designer.

A = the total shearing surface, in square inches, of the rivets (that is, twice the area of the rivet hole when a rivet is in double shear).

Only $\frac{15}{16}$ of the full area are to be taken when the riveting is done by hand.

### III. *Combined Rupture through Plate and Rivets.*

This case is only to be examined when the outer row has a wider pitch than the inner ones.

The formula to be applied in this case is :—

$$P=\frac{2(B\times R)+(C\times S)}{D\times l} \quad \ldots\ldots\ldots \quad \text{(III.)}$$

Where P R S D and $l$ have the same meaning as before.

B = the sectional area, in square inches, of the plate on the portion $l$ of the joint along the line of its supposed rupture, assuming that in case the plate is liable to corrosion, its thickness has been reduced by 0·04 inch.

C = the total area of the rivets which are supposed to shear on the length $l$, corrected, if required, in the same way as prescribed above.

For a rivet in double shear the resistance will be considered as being twice that of one in single shear.

### IV. *Rupture through Butt Straps.*

Rupture may take place along one of the inner rows of rivets (see P Q, figs. 1 and 2). The formulæ for this case, based on the same principle as (I.), are:—

$$\left.\begin{aligned} P &= \frac{2\,a\,R(t-0{\cdot}04)}{D} \\ \text{and}\quad t &= \frac{P\times D}{2\,a\,R}+0{\cdot}04 \text{ inch} \end{aligned}\right\} \quad . \quad . \quad . \qquad \text{(IV.)}$$

when plates are exposed to the direct action of products of combustion as in superheaters.

$$P=\frac{1{\cdot}6\,a\,R(t-0{\cdot}19)}{D},$$

when they are protected from direct action.

$$P=\frac{1{\cdot}8\,a\,R(t-0{\cdot}125)}{D},$$

where P D and R have the same meaning as before.

$t$ = thickness in inches, of butt strap, or sum of thicknesses, if there are two straps. (The thickness, of course, not to be less than required for caulking.)

$$a=\frac{q-d}{q},$$

where $q$ = pitch of rivets in the inner row, in inches.

$d$ = diameter in inches of the rivet holes in the inner row.

### V. *Combined Rupture through Butt Straps and Rivets.*

Formula (III.) applies to this case, B being the section of the butt strap or straps along which rupture would take place.

**Remarks.**—No rivet holes to be nearer the edge of any plate than the diameter of the rivet.

In zig-zag riveting, the distance between the rows is to be such that no rupture through plate or butt strap is to be feared along the zig-zag line.

When stays are bolted through the shell, they should be so arranged that they do not weaken the shell plates more than the riveted joints. If the resistance at the stay bolts is the smaller of the two, the plate's thickness shall be determined by it. It will be found from a formula the same as (I.), $p$ and $d$ applying to the stay bolts.

### FLAT PLATES.

The allowed working pressure or the thickness of flat plates is to be determined by the following formulæ:—

$$P=\frac{(t-1)^2}{a^2+b^2}\times\frac{T}{C},$$

and

$$t=1+\sqrt{(a^2+b^2)\frac{PC}{T}}.$$

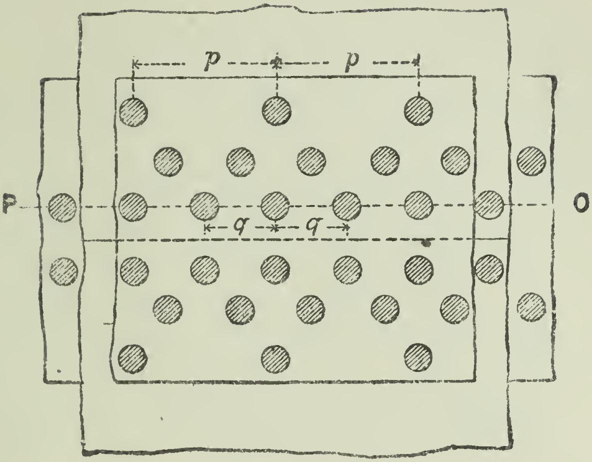


FIG. 1.

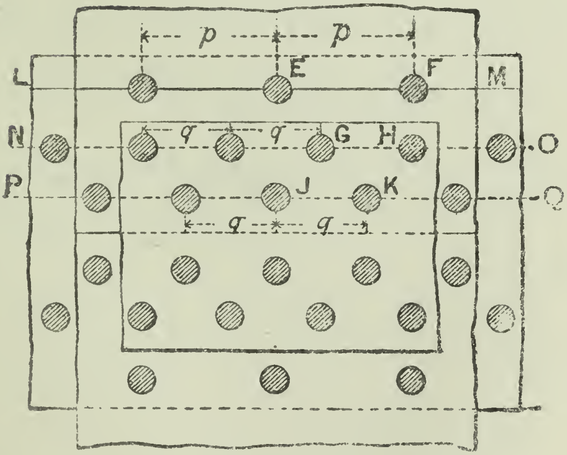


FIG. 2.

Where $P$ = allowed working pressure above atmosphere, in lbs. per square inch.

$t$ = thickness of plate in sixteenths of an inch.

$a$ = pitch of stays, in inches, in one row.

$b$ = distance, in inches, between two rows of stays.

*Note.*—When plates are effectively stiffened by doubling plates, well riveted thereto, and having a thickness $t_1$ in sixteenths, the value $(t + \frac{t_1}{2})$ may be substituted for $t$ in the formula.

In case of irregular staying, such as in the annexed sketch (fig. 3),

$$\tfrac{1}{4}(P_1 + P_2)^2$$

shall be taken instead of $a^2 + b^2$.

$T$ = tensile strength of the material in tons per sq. inch of the original section.

It is to be determined in the same way as for shell plates, that is :—

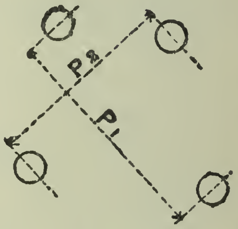


FIG. 3.

For steel it will be equal to the lower limit of the tensile stress, which is to be stated on the drawing.

$C$ = a constant, the value of which depends upon the mode of staying as follows :—

$C = 0·104$ when the stays are screwed into the plates, riveted over, and protected from hot gases; otherwise $C = 0·111$.

$C = 0·079$ when the stays are screwed into the plate and fitted with outside nuts at both ends, and protected from hot gases; otherwise $C = 0·084$.

$C = 0·062$ when the stays are fitted with inside and outside nuts and washers, provided the diameter of the outside washers be at least 0·4 of the distance between the rows of stays, and the thickness at least ⅔ that of the plate.

$C = 0·555$ when the stays are fitted with inside and outside nuts and washers, the outside washer being riveted to the plate and having ¾ of the plate's thickness and a diameter equal to 0·6 of the distance between the rows of stays.

$C = 0·050$ when the outside washers are replaced by strips of plate having a width of at least 0·6 of the distance between the rows of stays with a thickness not less than ¾ that of the plate; the strips being well riveted to the plate.

For the values of $\frac{T}{C}$, see following Table.

## VALUES OF $\frac{T}{C}$ IN THE FORMULA FOR FLAT PLATES.

| Tensile Strength of Plate in tons per square inch. | C=0·111. | C=0·104. | C=0·084. | C=0·079.* | C=0·062. | C=0·055. | C=0·050. |
|---|---|---|---|---|---|---|---|
| 22 | 198·2 | 211·5 | 261·8 | 278·5 | 354·8 | 400·0 | 440·0 |
| 23 | 201·3 | 221·1 | 273·7 | 291·1 | 371·0 | 418·2 | 460·0 |
| 24 | 216·4 | 230·8 | 285·6 | 303·8 | 387·1 | 436·4 | 480·0 |
| 25 | 225·5 | 240·4 | 297·5 | 316·4 | 403·2 | 454·5 | 500·0 |
| 26 | 234·6 | 250·0 | 309·4 | 329·1 | 419·4 | 472·7 | 520·0 |
| 27 | 243·7 | 259·6 | 321·3 | 341·8 | 435·5 | 490·9 | 540·0 |
| 28 | 252·8 | 269·2 | 333·2 | 354·4 | 451·6 | 509·1 | 560·0 |
| 29 | 261·9 | 278·8 | 345·1 | 367·1 | 467·7 | 527·3 | 580·0 |
| 30 | 270·0 | 288·5 | 357·0 | 379·7 | 483·9 | 545·5 | 600·0 |

* When plates are not subjected to flame or hot gases C may be reduced from 0·079 to 0·066; or the values of $\frac{T}{C}$ in this column may be multiplied by 1·2.

When the plates are in contact with steam on one side and flame or hot gases on the other, the thickness is to be increased.

For instance, when in return tube boilers, the top front plates are in no way protected from the hot gases, the working pressure or thickness will, in such a case, be determined by the formulæ:—

$$P = \frac{(t-2)^2}{a^2+b^2} \times \frac{0{\cdot}9\,T}{C}$$

and

$$t = 2 + \sqrt{(a^2+b^2)\,\frac{P\,C}{0{\cdot}9\,T}}.$$

When the said front plates are protected by a flame plate, n increase of thickness will be required.

When front plates are in two pieces, the lap should be double riveted if the thicker plate is ½ in. or above.

### STAYS.

The diameter of stays supporting flat surfaces is to be determined by the following formula:—

$$d = \tfrac{1}{8}\text{ inch} + \sqrt{\frac{Q}{300\,T}}.$$

Where $d$ = effective diameter in inches (for instance, the diameter at bottom of thread in screw stays).

Q = total load on stay in lbs.

T = tensile strength of the material, in tons per square inch.

For steel this tensile strength will be the lower limit chosen by the boiler designer; the actual strength may be applied if it is known from tests.

If the stays are not round, their cross section must be such that the stress per square inch, caused by the load Q, nowhere exceeds one 5·75th part of the tensile strength, after deducting $\frac{1}{16}$ of an inch all round as an allowance for corrosion or wear.

Welding of stays is not allowed.

For high working pressures, such as used in triple-expansion engines, it is recommended to screw all stays into the plates they support, in addition to fitting them with nuts.

This also applies to stay tubes, with the exception that it is recommended that nuts should not be fitted in combustion chambers.

## SCREWED STAYS.

The diameter of fire-box stays is to be determined by the following formulæ :—

$$d = 0{\cdot}177 + \sqrt{\frac{Q}{300\,T}}.$$

For margin stays the formula will be :—

$$d = 0{\cdot}177 + \sqrt{\frac{Q}{260\,T}},$$

$d$, Q, and T having the same meanings as above.

## VERTICAL BOILERS.

When the dished ends in vertical boilers are portions of spheres, their thickness must not be less than—

$$xt = \frac{Pr}{840} + 2.$$

When dished ends in steam chests are portions of spheres,

$$xt = \frac{Pr}{1140} + 3,$$

where $t$ = thickness of plate in sixteenths of an inch;
$x$ = percentage of joint;
P = working pressure in lbs. per square inch;
$r$ = radius of curvature in inches.

When the end is made in one plate, $x = 1$.

When the tops of furnaces are portions of spheres, the thickness must not be less than—

$$xt = \frac{Pr}{600} + 2,$$

where $x$, $t$, P, $r$ have the same meaning as before.

## FURNACES.

### *Plain Cylindrical Furnaces.*

When plain furnaces are made as truly circular as practicable, and of steel having a tensile strength of not less than 26 tons per square inch, the working pressure P and thickness of plate T may be calculated by the following formulæ :—

$$P = \frac{16{,}000T - 60L}{D},$$

$$T = \frac{P \times D + 60L}{16{,}000},$$

Where T = thickness of plate in inches.
P = working pressure in lbs. per square inch.
D = external diameter in inches.
L = length in inches.

*N.B.*—When the furnace is in one length L is measured from the centre of rivets at furnace mouth to those connecting back end to the tube plate, or to the commencement of flanging where back end of furnace is flanged. When the furnace is divided into two or more parts by Adamson joints L is measured to the centre of the joint.

For iron and low tensile steel use 14,400.

Furnace plates should not exceed $\frac{13}{16}$ inches in thickness.

### *Corrugated and Ribbed Furnaces.*

The plate thickness is to be found by the following formulæ :—

1. For corrugated furnaces :—

$$T = \frac{PD}{1260} + 2.$$

Where P = working pressure in lbs. per square inch.
T = thickness of plate in sixteenths of an inch.
D = outside diameter in inches measured on the top of the corrugations.

The formula applies to corrugations 6 inches long and $1\frac{1}{2}$ inch deep.

2. For ribbed furnaces, when manufactured to the satisfaction of the Administration :—

$$T = \frac{PD}{1160} + 2.$$

Where P and T are as above.

D = greatest outside diameter between the ribs, in inches.

The formula applies to ribs spaced 9 inches and projecting $1\frac{3}{8}$-inch, the difference between the greatest and the smallest diameters in any part of the furnace not exceeding $\frac{3}{1000}$.

3. For bulb furnaces :—

$$T=\frac{PD}{1260}+2.$$

Where D is the outside diameter in inches measured between the bulbs.

The co-efficients in the above formulæ apply to the case where the tensile strength of the material is 26 tons per sq. in. or above. When it is below 26 tons, the co-efficient is to be reduced $\frac{1}{26}$th for each ton below 26.

When the combustion chamber bottom is cylindrical thickness,

$$T=\frac{P\times 2\,R+60\,L}{14,400}.$$

### COMBUSTION CHAMBER GIRDERS.

The strength of girders on the tops of combustion chambers shall be determined as follows :—

$$P=\frac{Cd^2t}{(W-p)DL}\,.$$

Where P = working pressure in lbs. per sq. inch.
C = a constant, found as under.
N = number of bolts in each girder.
$d$ = depth of girder, in inches.
$t$ = thickness of girder, in inches.
$p$ = pitch of bolts in girder, in inches.
L = length of girder between supports, in inches.
D = distance between girders, centre to centre, in inches.
W = width of firebox from tube plate to back plate, in inches.

$$C=\frac{13,200N}{N+1}\text{ for odd numbers of bolts.}$$

$$C=\frac{13,200(N+1)}{N+2}\text{ for even numbers of bolts.}$$

## CONSTRUCTION OF ENGINES.

## Shafts for Screw Steamers.

### A. CRANK SHAFTS.

When the crank of a screw engine is not overhung, the diameter of the shaft shall be determined by one of the following formulæ :—

*For non-compound condensing engines :*

$$d=\sqrt[3]{\frac{nPLD^2}{C}}\quad .\ .\ .\ .\ .\ .\ .\ .\ .\ .\quad (A)$$

*For double, triple, and quadruple expansion engines :*

$$d=\sqrt[3]{\frac{PL(n_1D_1^2+0.1\,n\,D^2)}{C}}\quad .\ .\ .\quad (B)$$

For shafts having a single overhung crank, the form under the radical sign is to be multiplied by

$$s+\sqrt{s^2+1}.$$

For two cylinder single crank tandem engines the formula will therefore be :—

$$d=\sqrt[3]{\frac{\mathrm{P\,L}\,(\mathrm{D}_1^2+0.1\,\mathrm{D}^2)\,(s+\sqrt{s^2+1})}{\mathrm{C}}}\quad . \quad (\mathrm{C})$$

In those formulæ :

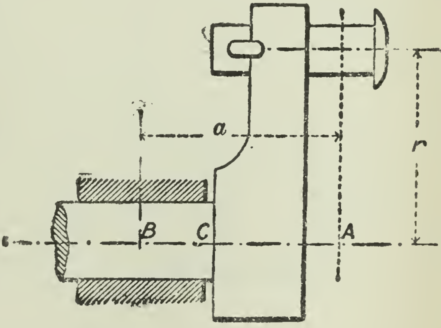
$d$ = diameter of the after shaft bearing in inches.
$n_1$ = number of high pressure cylinders.
$\mathrm{D}_1$ = diameter of each high pressure cylinder in inches. If there are several high pressure cylinders the diameters of which are not the same, $n_1\,\mathrm{D}_1^2$ represents the sum of the squares of their respective diameters.

$n$ = number of low pressure cylinders.
D = diameter of each low pressure cylinder in inches. If there are several low pressure cylinders the diameters of which are not the same $n\,\mathrm{D}^2$ represents the sum of the squares of their respective diameters.

*N.B.*—For triple or quadruple expansion engines the intermediate cylinders do not come into account in the formulæ.

L = length of stroke in inches, common to all pistons.
P = boiler pressure above atmosphere in lbs. per sq. inch.

$s=\frac{a}{r}$ (see below). In order to determine $a$, B is supposed to be situated half-way the length of the bearing, unless the latter be longer than $1\frac{1}{2}$ times the diameter ; in this case B C may be considered as being equal to $\frac{3}{4}$ of the diameter.



C = a constant, the values of which are given below for certain cases.

If the diameter of shaft be above 15″, it should be increased by an amount to be determined by the Administration ; for built-up shafts, however, this latter increase will not be required.

For hollow shafts, the diameter must be increased by

1 per cent. if the diameter of the hole is 0·4 of the outside diameter ;
2 ,, ,, ,, 0·5 ,,
5 ,, ,, ,, 0·6 ,,
10 ,, ,, ,, 0·7 ,,

If the hole is under 0·4 of the outside diameter, no increase will be required.

The Administration may allow a reduction on the diameter in certain special cases, for instance in well-balanced engines with light moving parts or for very superior workmanship, etc. On the other hand, the Administration may require an augmentation for engines which differ much from the average proportions found in practice, thus, for instance, for engines having a comparatively small stroke; for compound engines the low pressure cylinder of which has a very large size compared with the high pressure cylinder, etc.

### Values of C in Crankshaft Formulæ (A), (B), and (C).

| Description of Engine. | Cranks. | | No. of Cylinders. | No. of H. P. Cylinders. | No. of L. P. Cylinders. | Constant C. | Formula. |
|---|---|---|---|---|---|---|---|
| | No. | Angle. | | | | | |
| Simple or non-compound. | 2 | 90°<br>100°<br>120°<br>140°<br>160–180° | 2 | ... | 2 | 4800<br>4370<br>3790<br>3450<br>3360 | A |
| Two cylinder compound. | 2 | 90°<br>100°<br>120°<br>140°<br>160–180° | 2 | 1 | 1 | 3400<br>3090<br>2680<br>2440<br>2380 | B |
| Triple expansion. | 3 | 120° | 3 | 1 | 1 | 3900 | B |
| Four cylinder triple or quadruple. | 4 | 90°<br>Angles arranged to reduce maximum torsion moment. | 4 | 1 | 2 or 1 | 4000<br>Not to exceed 4100 | B |

### Propeller, Tunnel and Thrust Shafts.

The diameter of propeller shaft is to be found from the following formula:—

$$\delta = d + \left(1{\cdot}7\frac{D}{d} - 15\right) \times \frac{d}{100}.$$

Where $\delta$ = diameter of propeller shaft;
$D$ = diameter of propeller; and
$d$ = diameter of the crank shaft calculated from one of the formulæ (A), (B), or (C), all in inches.

It is recommended to fit the propeller shaft in such a way that it cannot move endways, if for some reason or other it has been uncoupled from the rest of the shafting.

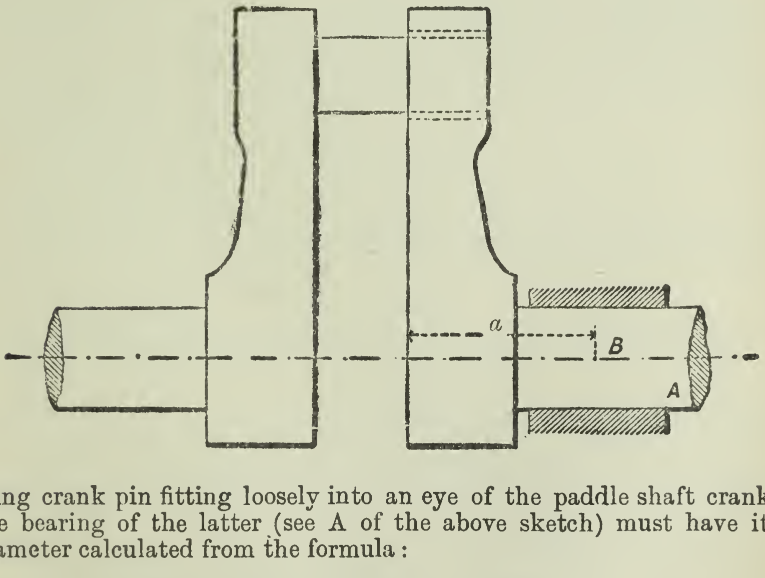
Liners fitted on propeller shafts to be tapered off at ends.

For tunnel shafts a reduction of 6 per cent. on the diameter of the crank shaft will be allowed.

The diameter of thrust shaft at the bottom of the collars, both between and immediately beyond these latter, to be equal to that of the crank shaft, and tapered off at each end to the smaller diameter of the body of the shaft. The thrust of the screw propeller must be taken up by an efficient thrust block, so as to prevent any fore-and-aft strain on the crank shaft.

## Shafts for Paddle Steamers.

In side wheel steamers having double, triple, or quadruple expansion engines with an intermediate shaft, each end of which carries an overhung crank pin fitting loosely into an eye of the paddle shaft crank, the bearing of the latter (see A of the above sketch) must have its diameter calculated from the formula:



$$d = \sqrt[3]{\frac{P\,L\,(n_1\,D_1^2 + 0{\cdot}1\,n\,D^2)\,(s + \sqrt{s^2+1})}{C}} \quad . \quad . \quad (D)$$

where the letters have the same meaning as before, except that $a$, for determining $s=\frac{a}{r}$, is to be measured as shown in the above ketch, the point B being the middle of the bearing.

For two cylinder compound receiver engines with two cranks at 90°

C = 13,000 for navigation in smooth water.*
= 7,100 for coasting vessels.
= 5,700 for sea-going vessels.

For triple expansion engines with three cylinders and three cranks at 120° :—

C = 14,900 for navigation in smooth waters.*
= 8,150 for coasting vessels.
= 6,540 for sea-going vessels.

The diameter of the outer bearing of the paddle shaft and of the intermediate shaft to be submitted to the Administration or the Surveyors for approval. The same applies to other cases not dealt with in this paragraph.

## Shafts for Turbine Engines.

In turbine engines, where I.H.P. is the estimated power transmitted by each shaft,

$d$ = diameter of tunnel shafting in inches.
$d$ = diameter of propeller shaft in inches.
D = diameter of propeller in inches.
R = number of revolutions per minute.

$d$ and $d_1$ should be given by the following formulæ :

$$d=\sqrt[3]{\frac{70\times \text{S.H.P.}}{\text{R}}},$$

$$d_1=d+\frac{\text{D}}{160},$$

and rotor shaft is to have a diameter at the smallest part at least 5 per cent. greater than the tunnel shafting.

## STEAM PIPES.

Solid-drawn copper pipes to be of thickness $t=\frac{\text{PD}}{6400}+0{\cdot}05$.

Brazed copper pipes, $t=\frac{\text{PD}}{5440}+0{\cdot}05$.

Steel pipes to be of thickness, solid drawn, $t=\frac{\text{PD}}{7885}+0{\cdot}08$.

If welded, $t=\frac{\text{PD}}{6850}$.

If riveted, $t=\frac{\alpha\text{PD}}{7885}+0{\cdot}08$, $\alpha$ being the strength of joint.

---

* Not now admitted.

P is boiler pressure in lbs. per square inch, and D is inside diameter in inches.

**Spare Gear for Engines and Boilers.**—For vessels employed in Mediterranean, in Continental coasting trades, and making long voyages, the following articles of spare gear are required to be on board:—

2 Top end bolts and nuts.
2 Bottom end bolts and nuts.
2 Main bearing bolts and nuts.
1 Set shaft coupling bolts and nuts (for one coupling).
½ Set feed pump valves.
½ Set bilge pump valves.
1 Set piston rings.
1 Set safety-valve springs.
5% of total number boiler tubes.
4% of total number condenser tubes.
½ Set of fire bars.
1 Canvas fire hose.
1 Set of spanners.
Plugs for boiler tubes.
12 Gauge glasses.
1 Steam gauge.

For vessels under French flag the following *additional* articles are required:—

1 Pair crank-pin brasses (or sufficient white metal and means for applying same).
½ Set bilge pump valves (metal).
1½ Set bilge pump valves (india-rubber).
½ Set bilge pump valve seats (if removable).
2 Sets gauge glasses.
½ Set fire bars.
1 Set electric hand lamps.
1 Safety lamp.
1 Set firing tools.
Also in vessels carrying coal or other dangerous cargo:—
4 Safety lamps.

And in vessels fitted with electric machinery, the necessary articles of spare gear according to the type of apparatus employed.

A further list is given in the Rules of articles that are *recommended* to be carried on board, and when these articles are so carried the vessel will be entered in the Register with the special mark ( ).

**Qualities and Testing of Materials.**—All material used in vessels for which the Special Survey mark is required must be tested, at the manufacturers' works if possible, and in the presence of a Surveyor to the Bureau Veritas, as hereinafter described, and all material that

## Steel for Boilers.

| ITEMS. | TENSILE AND ELONGATION TESTS. | | | | | BENDING TESTS. | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | Dimensions of Test Pieces. | Tensile Strength, Tons per sq. inch. | | Elongation. | | Cold Tests. | | | Quenching Tests. | | |
| | | Minimum. | Maximum. | For Tensile Strength of Tons. | Percentage in 8 ins. | Dimensions of Test Piece. | Internal Radius. | Angle. | Dimensions of Test Piece. | Internal Radius. | Angle. |
| Shell plates. | Working length 8 ins.<br>Finished breadth 1¼ ins. | 26 | 32 | 26–32 | 24% to 20% | Strips 2 ins. wide. | Radius equal to thickness of plate. | 180° | Strips 2 ins. wide. | Radius equal to 1½ times thickness of plate. | 180° |
| | | | | 29 | 22% | | | | | | |
| End plates. | | 25·5 | 30·5 | 30·5 | 20% | | | 180° | | | 180° |
| | | | | 25·5 | 25% | | | | | | |
| Furnaces and Fire-boxes. | | 24 | 29 | 29 | 21% | | | 180° | | | 180° |
| | | | | 25 | 25% | | | | | | |
| | | | | 24 | 26% | | | | | | |

Steel for shell plates may be up to 35 tons tensile if the elongation is not less than 10 per cent.

## Steel for Boilers (*continued*).

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Long stays and Fire-box stays. | Working length 8 ins. or 4 ins. Diameter turned to $\frac{9}{16}$ in. | 25·5 | 30·5 | 30 | In 8 ins. 20% | In 4 ins. 25% | Short length of bar. | Radius equal to diameter of bar. | 180° | Short length of bar. | Radius equal to $1\frac{1}{2}$ times diameter of bar. | 180° |
| | | | | 29 | 21% | 25% | | | | | | |
| | | | | 28 | 22% | 26% | | | | | | |
| | | | | 27 | 23% | 27% | | | | | | |
| | | | | 26 | 24% | 27·5% | | | | | | |
| | | | | 25 | 24·5% | 28% | | | | | | |
| Rivets. | Working length 8 ins. | 24 | 29 | 29 | 25% | | Short length of bar. | Doubled over. | 180° | Short length of bar. | Doubled over. | 180° |
| | | | | 28 | 26% | | | | | | | |
| | | | | 27 | 27% | | | | | | | |
| | | | | 26 | 27·5% | | | | | | | |
| | | | | 25 | 28% | | | | | | | |
| | | | | 24 | 29% | | | | | | | |

Quenching tests to be made by heating the test pieces to a dull red and then quenching them in water at about 82° F.

A complete set of tests is to be made for each rolled plate and also for the long stays and for the fire-box stays of each boiler.

At least two sets of the following tests also to be made on the rivets of each boiler:—Cut short length equal to two diameters from rivet bar and compress cold to half length; cut similar piece, heat to dull red, and compress to quarter length, after which punch the pieces through with a punch having a diameter equal to $\frac{3}{4}$ that of the rivet.

## Steel Forgings and Castings, Cast-iron, Copper, Bronze, etc.

| ITEMS. | TENSILE AND ELONGATION TESTS. | | | | | COLD BENDING TESTS. | | | REMARKS. |
|---|---|---|---|---|---|---|---|---|---|
| | Dimensions of Test Pieces. | Tensile Strengths. Tons per sq. in. | | Elongation. | | Dimensions of Test Pieces. | Internal Radius. | Angle. | |
| | | Minimum. | Maximum. | For Tensile of Tons. | Percentage. | | | | |
| Steel forgings. | Working length 4 ins. diameter turned to $\frac{9}{16}$ in. | 27 | 32 | 27 | 26% | 1 in. square and 10 ins. long. | $\frac{3}{4}$ in. | 180° | Two tests to be made for any number of forgings from the same charge. Test pieces to be cut cold from forging after annealing, and as far as possible from end of ingot at which metal was run in. |
| | | | | 28·5 | 25% | | | | |
| | | | | 30 | 24% | | | | |
| | | | | 32 | 23% | | | | |
| Nickel steel forgings, 3·25% nickel, 0·35% carbon. | Do. | 37 | 42 | 37–42 | 20% | Instead of bending test a piece 1 in. sq. placed on knife edges $6\frac{1}{4}$ ins. apart must stand 15 blows from weight of 40 lbs. falling 9 ft. | | | Test pieces to be taken (one for each piece) after forgings have been annealed. |
| Steel castings. | Do. | Generally<br>28 | 38 | 28 | 16% | 1 in. square | $1\frac{1}{2}$ in. | 90° | Test piece 1 in. square placed on knife edges $6\frac{1}{4}$ ins. apart must stand without fracture a series of shocks from a 40 lb. weight falling $3\frac{1}{2}$ ft. to 5 ft. by 2 in. increments after each blow. One test from every charge and one for every large casting. |
| | | Engine parts subject to alternating stresses<br>28 | 35 | 31 | 14% | | | 90° | |
| | | Crankshafts<br>28 | 30 | 35 | 12% | | | 90° | |
| | | | | 38 | 10% | | | 90° | |

## Steel Forgings, etc. (*continued*).

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Cast iron. | Diameter turned to ⅝ in. | 7½ (common)<br>11 (superior) | | .. | .. | .. | .. | .. | Pieces 1⅝ ins. square placed on knife edges 6¼ ins. apart must stand without fracture a series of blows from a 27-lb. weight beginning at 10 ins. and advancing by 2 ins. to 16 ins. for common iron, and at 12 ins. and advancing by 2 ins. up to 18 ins. for superior iron. |
| Copper plates. | Working length 8 ins., finished breadth ¾ in. | 12·5 | .. | .. | 22% | strips 6 ins. long and 2 to 2½ ins. wide. | 1½ times thickness of plate. | 180° | Hot bending tests on 6-in. × 2-in. to 2½-in. strips must show that plates will bend double with faces touching without any sign of fracture. |
| Brass plates. | | 19 | .. | .. | 12% | | | | |
| Copper pipes. | Working length 8 ins., finished breadth ¾ in. | 13 | .. | .. | 30% | .. | .. | .. | End of pipe to be flanged over to ⅙ external dia. of pipe, but not exceeding 1¼ ins. Radius of bend equal to thickness of tube: a piece of pipe 4 ins. long to be sawn through lengthwise and bent back reverse way to form a cylinder. Internal pressure test 430 lbs. per sq. in. Test pieces to be annealed. |
| Brass tubes. | .. | . | .. | .. | .. | . | .. | .. | Flanging test as for copper pipes (above). Diameter of tube to be increased ⅜ in. by drift. Piece of tube to be completely flattened. Hydraulic test 430 lbs. per sq. in. Test pieces to be annealed. Condenser tubes to be completely flattened. Hydraulic test 570 lbs. per sq. in. Test pieces to be annealed. |
| Turbine blades 72 Cu + 28 Zn. | . | 27·5 | .. | .. | 20 | ⁹⁄₁₆ diam. × 4 ins. long. | .. | .. | .. |

satisfactorily withstands these tests is to be legibly stamped by the Surveyor. If any test of steel is unsatisfactory the Surveyor is entitled to condemn all material from that charge.

Steel must be made exclusively by the Siemens-Martin process; nevertheless steel produced by the electric furnace may be adopted with the sanction of the Administration.

Copies of lists of material giving all needful particulars are to be handed to the Surveyors.

In the United Kingdom, the tests of the British Standards Committee will be accepted if carried out in the presence of a Bureau Veritas Surveyor.

**Boiler Tubes.**—Material to have an ultimate strength lying between 22 and 25 tons per square inch, and an elongation of 22% on 8 inches if thicker than ⅛ inch, and 20% if thinner. Test pieces to be 1¼ inches wide.

Material also to stand (cold) the following tests without cracking:—

1. End of tube to be expanded by a drift to 8% increase of outside diameter if over $\frac{3}{16}$ inch thick, and to 10% for thinner tubes.

2. A piece 4 inches long to be sawn through lengthwise at a reasonable distance from the weld, opened and rolled again in the opposite way to a cylinder.

3. A piece 4 inches long to be flattened, the weld being in the fold, until the distance between the sides is equal to the thickness for tubes of or over $\frac{3}{16}$ inch thick, and until the sides are close together for thinner tubes.

4. The end of the tube to be flanged at right angles into a rim of four times the thickness for tubes under $\frac{5}{32}$ inch thick and of ⅝ inch for thicker tubes.

5. Hydraulic test to 560 lbs. per square inch, and each tube to be hammered whilst under pressure.

For brass castings the minimum strength is to be 11·5 tons per square inch, with an elongation of 8 per cent. on 4 inches.

For high tensile bronze the minimum strength is to be 28·5 tons per square inch, with an elongation of 18 per cent. on 4 inches.

In both cases the test pieces are to be turned to $\frac{9}{16}$ inch diameter.

## Electric Lighting.

§ 1. *Dynamos* to be of an approved type; continuous currents are to be preferred. The engines should be fitted with a governor and situated in the engine-room or in a separate adjacent compartment and efficiently ventilated.

The dynamo circuits to be tested to 800 volts for 5 minutes.

§ 2. *Switch-Boards.*—The fittings on main switch-board should be mounted on slate or other incombustible material, placed near the dynamos, and be accessible from the back, if possible, unless all connections be established on the front of the switch-board. From this the main circuits should be led to auxiliary switch-boards for distributing the current to the various branches which may not be taken directly off a main cable.

A volt-meter for each installation and an ampère-meter for each dynamo to be supplied.

§ 3. *Leads* to be of copper wire having a conductivity of at least 98 per cent. Single wires should not be less than 20 S.W.G., with a sectional area of at least 1 square inch per 1250 ampères. The insulation on the leads should be absolutely watertight and be capable of bearing a temperature of 150° F. without being softened. All cables should have an insulation resistance of not less than 700 megohms per statute mile after twenty-four hours' immersion in sea water at a temperature of 60° F.

If alternating currents are used the insulation must be double that required for a continuous current of same voltage.

It is recommended to test the insulation of the electric plant before proceeding on a long voyage.

Leads should be accessible. In cabins they should be laid in battens with covers screwed on; where exposed to moisture they should be lead covered, and be armoured or protected by galvanized iron casings in cargo holds or wherever liable to be injured.

Watertight packings of an approved type must be fitted where leads pass through decks or watertight bulkheads, and the leads should be carefully protected from chafing against holes in beams, &c. All bends should be as easy as possible.

§ 4. *Joints* must be carefully made and insulated to the same degree as the cables, and should be in places always accessible.

Resin should be employed as a flux for soldering.

In the double-wire system joints in flow and return wires should not be opposite one another.

In single-wire plants the joints with hull must be accessible. Large cables should be secured in a copper plate bolted to the steel-work of the hull, which should be scraped bright at the contact, the area of contact being at least five times the sectional area of cables. A brass screw with non-oxidizable washer may be employed for single lamps and small cables.

§ 5. *Switches* should be made so as to act quickly, and be either full on or off without remaining in an intermediate position.

They should have large rubbing surfaces, and be so arranged that the friction takes off the oxide formed. A switch should be fitted to each main and branch circuit, and the more important ones arranged so as not to be tampered with by irresponsible persons. In places affected by moisture they should be fitted in watertight boxes having portable covers.

§ 6. *Fusible Cut-outs* should be fitted as a rule at the origin of each branch circuit, and be situated close to switch, those for the larger cables being placed on the switch-board. In double wire plants a cut-out should be fitted at the origin of each wire of each circuit. Like switches they should be placed in accessible positions and arranged so as not to be tampered with, be mounted on an incombustible base, and be fitted with a strong incombustible cover, which should be watertight where exposed to moisture. They must melt with a current double the normal one, *i.e.* at 2500 ampères per square inch, and care must be taken to avoid mistakes in the size of fuses.

§ 7. *Lamps* to be of an approved pattern, strong, well insulated, and efficiently secured in their sockets.

Those in machinery space must be watertight and be protected by a glass globe with wire-netting.

In crew spaces, passages, holds, &c., they must be of a strong pattern efficiently protected by wire-guards. Deck lamps as well as side and masthead lamps must be perfectly watertight and be fitted with detachable connections.

§ 8. In Oil Vessels alternating current dynamos are strictly prohibited, and single-wire plants should not be fitted.

The insulating material must be such as not to be injured by oil or vapours.

The leads must not be run through the tanks, and no switches, cut-outs or joints may be placed in pump room.

All parts of circuits or fittings where sparks might be produced must be above the tank-deck, and situated where no gas can accumulate.

Arc lamps may not be used, and all incandescent deck and hold lamps are to be strongly protected by air-tight globes with wire-netting.

§ 9. *Compasses.*—The following precautions are recommended in order to lessen the deflections produced by electric-currents: compasses, especially the standard compass, should be at least 33 feet from any continuous current dynamo or electromotor, and 50 feet from alternating current machines. No single wire to be nearer a compass than 16 feet, and if this be not possible, the double wire or concentric system should be adopted in the vicinity of the compass. Chronometers also should be kept at a fair distance from dynamos and leads.

The influence of electric currents on compasses must be tested when these are being adjusted, with the vessel's head in any position and with all possible arrangements of current in the leads likely to affect the compass.

# APPENDIX C.

## RULES OF THE GERMAN GOVERNMENT FOR THE MERCANTILE MARINE.

### Regulations respecting Steam Boilers.

1. *Plating of Boilers.*—When the smallest dimensions of a cylindrical boiler exceeds 25 centimetres, or of a spherical-shaped boiler 30 centimetres, the portions of the boiler exposed to the flame, furnaces, and tubes shall *not* be made of cast iron.

Brass (copper) plates are only to be used for furnaces where their smallest dimension does not exceed 10 centimetres.

2. *Furnaces, Flues, and Combustion Chambers.*—Flues running round or through steam boilers must, at their highest point, be at least 10 centimetres below the minimum water level. This minimum level must, in lake and river vessels, be maintained when the vessel is inclined at an angle of 4°, and for sea-going ships at an angle of 8°. In boilers with a breadth of 1 to 2 metres, the distance to the water level must be at least 15 centimetres, and in boilers of greater breadth at least 25 centimetres.

These conditions do not apply to boilers in which the tubes are less than 10 centimetres diameter, nor to those wherein the portions of the plates in contact with steam are not liable to become red hot. The risk of plates becoming red hot may be assumed to be done away with if the heating surface exposed to the flame, before the flame reaches the point in question, is, with natural draught 20 times, and with forced draught 40 times, greater than the area of the grate. The highest point in marine boiler flues, etc., is the *upper* surface of the plates in question.

3. *Feed Apparatus.*—Each boiler must have a feed valve which will close by boiler pressure when the feed is off. Where only one feed check-valve is provided shut-off valves must be fitted in the feed delivery pipes.

4. Each boiler must be fitted with two independent feed apparatus, each worked separately, and each to be capable, by itself, of keeping the boiler supplied. Several boilers connected together and used for the same purpose may, in this respect, be regarded as one boiler. Each feed apparatus to be capable of delivering 30 litres of water per sq. metre of heating surface per hour. Apparatus for boilers worked under forced draught, or having heating surface less than 35 times the grate area, must be proportionately larger. Each feed apparatus must have an independent suction pipe, or else a suitable arrangement of

switch shut-off valves must be fitted. An auxiliary feed apparatus fo a main boiler may be used for feeding a donkey boiler if there are independent delivery pipes, or if suitable switch shut-off valves are provided. If an injector is used for auxiliary feed purposes it must be capable of working satisfactorily with both high and low pressure steam; if it will not so work, some other provision must be made for use when pressure is low. Where a main boiler is used for driving winches in port, the main engine feed pumps will not be reckoned as a secondary feed apparatus. If supplementary feed arrangements, beyond those required by the regulations, are provided, the piping for them must either be independent of that belonging to the regulation apparatus, or must be capable of being shut off from the latter by valves. Each main feed-pump must be fitted with an escape valve that cannot be shut off from the pump.

5. *Water Gauges.*—Each boiler must be fitted with a gauge glass, and also with a second means of ascertaining the height of the water. Each of these fittings must have a separate connection to the boiler, or if they have a common connection, the stand pipe must be at least 60 square centimetres in area, or 88 millimetres in diameter. Pipes for connecting water-gauges to boilers must not be less than 16 centimetres in area, or say 45 millimetres in diameter. Long or much bent pipes must be larger in proportion. Internal connecting pipes must not be used, but two pipes may be led to one opening in the plating, provided the area of opening be made equal to the combined areas of the pipes. Gauge glasses are to be placed so as to be easily visible. The mark indicating lowest water-level should not be higher than half-glass, and the mark showing highest point of heating surface should be above upper edge of lower nut which secures glass. The cocks must be easy to open and shut, and it must be possible to replace a broken glass without risk to the operator.

6. Water test cocks to be fitted; the lowest one must be placed at the level of the regulation water-line. All gauge cocks must be so fitted as to have a straight way through for cleaning them from scale or salt. Spindle valves must not be used.

7. *Water Level.*—The regulation lowest water level is to be distinctly marked on the gauge glass, and also in a prominent place on the boiler shell. The level of the flue, at its highest point, is also to be plainly and permanently marked on the shell of the boiler in way of the ship's beam.

Two water-gauge glasses are to be fitted on the boiler at the standard level of the water when the vessel is inclined as above, one on each side of the boiler, as far apart as possible, and symmetrically placed right and left of centre line of boiler.

When these two gauge glasses are fitted, the additional means named in Rule 5, for ascertaining the water level, may be dispensed with.

The marks indicating lowest water level are to be not less than 5 centimetres long and are to be cut into the plating near to the water-gauges. They are also to be distinguished by having the letters N. W. stamped on them. They must be on before the inspection of the boiler is made, and must be visible after it is lagged. Where water-gauges of main boilers of river steamers and barges and of donkey boilers are direct on the boiler shell, a plate must be fitted at the lowest water-level mark with "Niedrigster Wasserstand" on it, and an arrow pointing to the glass. Where water-gauges are connected to boilers by pipes (as must be the arrangement in all sea-going steamships) a similarly marked plate is to be fixed to the stand-pipe. The highest point of the heating surface is to be indicated as follows:—Where the water-gauge is fitted direct to the boiler shell a plate indicating the point in question, and marked "Hochster Feuer berührter Punkt," is to be fixed near the gauge. Where the gauge is connected to the boiler by pipes a similar plate is to be affixed to the gauge standard. These plates are to be permanently secured—not with removable screws.

In cases where the fore and aft trim of the vessel is subject to considerable variation, the minimum water level (paragraph 2) must either be raised 10 centimetres, or the mark indicating highest point of heating surface must be raised a similar amount. The permanent mark showing regulation lowest water level must, however, correspond with the particulars given on the maker's name plate, as hereafter specified.

A third water-gauge must be fitted on every double-ended boiler.

8. *Safety-Valves.*—Each boiler must be fitted with, at the least, one reliable safety-valve.

When several boilers have a common steam chest, from which they cannot be separately disconnected, two safety-valves will be sufficient.

Boilers of steamships, locomotives, and portable engines must have at least two safety-valves.

In steamships, except those that are sea-going, one valve is to be placed in such a position that the load on it can easily be ascertained from the deck.

The valves must be so arranged that they can be readily lifted at any time. They are to be loaded so that they will blow off immediately the working pressure is reached.

Safety-valve areas must be such that after firing up for 15 minutes the pressure does not accumulate more than 10 per cent.

Provided that the springs are satisfactorily fitted and loaded, the area of safety-valves, in square millimetres, may be as follows, for each square metre of heating surface:—

| Pressure above atmosphere, | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Area in sq. millimetres, | 131 | 112 | 98 | 86 | 79 | 72 | 66 | 60 | 56 | 54 | 52 | 51 |

For boilers worked under forced draught, or in which the heating surface is less than 35 times the grate surface, the safety-valve areas must be correspondingly increased. Safety-valve boxes must be fitted with drain pipes that cannot be shut off. The waste steam pipes from all safety-valves must blow off into the open air. Blowing off into a tank, or into the chimney, will not be permitted. The thickness of the washers which determine the compressions of the safety-valve springs will be fixed by the Official Surveyor and recorded in his book, and no alterations must be made except by his authority. Exception is made, however (by the regulations of 19th December 1883), in the case of long voyages, and the chief engineer of the vessel is permitted to re-adjust the washers in case the valves blow off too freely. To meet this case the washers may be fitted in halves.

9. *Pressure Gauge.*—Each boiler must have a reliable pressure gauge on which the highest regulation working pressure is to be plainly marked.

Boilers of steamships must have two pressure gauges—one clearly visible to the fireman, the other, except when the vessel is sea-going, in a convenient position for being observed from the deck. If a vessel has several boilers connected to a common steam chest, one pressure gauge on deck will be sufficient, in addition to the one on each boiler.

Gauges to be placed where they can easily be seen. Dials to be marked in kilogrammes per square centimetre (or in atmospheres,—1 atmosphere equals 1 kilogramme per square centimetre), and marking must go up to 50 per cent. above the regulation working pressure. In vessels having more than one boiler, one pressure gauge (in a prominent position in the engine room), in addition to those on the boilers, will suffice. The pipe to this gauge must be connected to all boilers, but must be capable of being shut off from each.

10. *Name Plate on Boiler.*—(1) Each boiler must have the highest working pressure, the maker's name, the shop number of the boiler, and the date of completion, plainly and permanently stamped on it. Steamships, moreover, must also have the standard lowest water level marked on.

(2) Boilers that have already been constructed, at the date of these regulations, are not required to be altered to comply with them.

(3) The regulations for boilers of steamships apply in all cases in which boilers are permanently fixed in, or connected with, the vessel.

The address of the maker must be given as well as his name.

10*a*. These particulars must be stamped on a metal plate, which is to be fixed on the boiler by copper rivets, the heads of which are to be not less than 12 millimetres in diameter. The plate to be in such a position that it is easily visible when there are casings on the boiler.

11. *Testing of Boilers.*—Every new boiler must, on completion, be tested by water pressure.

The test pressure for boilers intended for a working pressure of not more than 5 atmospheres to be double the working pressure, and for boilers intended for a working pressure of more than 5 atmospheres, the test pressure is to be 5 atmospheres above the working pressure.

For boilers where the working pressure is under atmospheric pressure, the test pressure is to be 1 kilogramme per square centimetre. The plates of the boiler must withstand the test pressure without showing any permanent change of form, and must also remain tight under the pressure.

The boiler is to be regarded as leaky, if the water at the highest pressure leaks through in other forms than dewy moisture or pearly drops.

11*a*. When the boilers have been tested and found satisfactory, the copper rivets which fix the brass plate are to be officially stamped, and an impression of this mark is to be inserted on the certificate of test.

12. When boilers have undergone a thorough repair in the boiler shop, or when they have had to be stripped and extensively repaired on board, they must be tested by hydraulic pressure the same as new boilers.

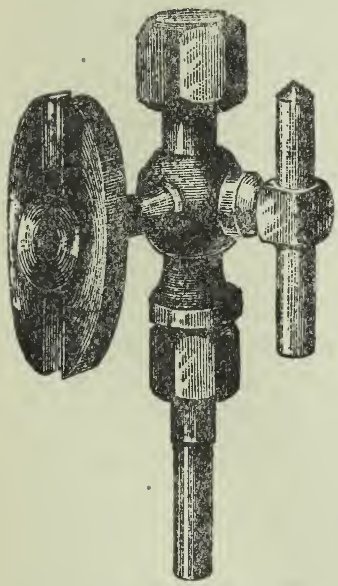
When in boilers with internal furnace, the furnace has been taken out to repair or renew, and when in locomotives the fire-box has been taken out to repair or renew, or when in cylindrical boilers one or more plates have been renewed, the boilers must, on completion of such repairs, be tested by hydraulic pressure as above.

When a boiler has undergone considerable repairs the steam pressure to be allowed must be determined afresh.

13. *Test Gauge.*—The test pressure is to be registered by an open mercurial gauge, or by the standard pressure gauge provided by the inspecting official.

Each boiler to be tested must be fitted with a connection to suit the standard gauge, and by which the gauge may be applied by the inspecting officer.

To facilitate application of official test gauge, a flanged branch is to be provided immediately below the usual pressure gauge. The opening in the flange and cock must not be less than 7 millimetres, the length of the branch must be 60 millimetres, and its breadth 25 millimetres. Form to be as shown in sketch.



## Rules for Strength of Boilers.

*5th August* 1890.

The quantities are expressed in centimetres and kilogrammes:—
(1 kilo. per sq. cm. = 1 atm. = 14·223 lbs. per sq. in. English).

Formulæ for thickness of shell plate and working pressure:—

$$\text{Working pressure} = \frac{S \times R \times 2T}{D \times F}.$$

$$\text{Thickness of plate} = \frac{P \times D \times F}{S \times R \times 2}.$$

Where P = highest steam pressure (kilos. per sq. cm.).
S = lowest tensile strength of material.
T = thickness of shell plate.
D = greatest internal diameter of boiler.

F = { 4·75 when rivet holes in longitudinal seams are drilled and rivets closed by hydraulic pressure.
4·5 when longitudinal seams are double strapped and have holes drilled and rivets closed by hydraulic pressure.

$$R = \frac{\text{Pitch} - \text{Diameter of rivet}}{\text{Pitch}}.$$

$$\text{or } R = \frac{\text{Area of rivet} \times \text{No. of rivets in pitch} \times a}{\text{pitch} \times \text{thickness of plate}} \times 0{\cdot}9 \times \frac{S_1}{S}.$$

Where $a$ = { 1 for lap and single riveted butt joints.
2 for double riveted, etc., butt joints.

$S_1$ = Tensile strength of rivet material. Unless actual test has been made, this is to be taken as 3300 for iron, and 3800 for steel; but it must not exceed 4800 kilos. per sq. cm. in any case.

The lower of these two values of R must be used.

S is to be taken as 3300 for iron; for steel the lowest value given by the tests is to be used, but this must not exceed 5000 kilos. per sq. cm., and the material must show an elongation of at least 20 per cent. in a length of 20 centimetres. Steel of higher tensile strength may only be used under specially arranged conditions as to tests, etc.

In riveted joints the shearing strength of the rivets must not be less than the tensile strength of the plate left between holes.

Butt-straps are to be of the same quality as the shell plates, and of not less than 75 per cent. of their thickness.

In boilers without a central circumferential seam the outside butt-strap must not be thinner than the shell plate. If the shell plates are thicker than 1·25 centimetres, the central circumferential seam must be double riveted, and if they are 2·5 centimetres or more it must be treble riveted.

Rivet holes must be at least one diameter of rivet from edge of plate. Diameter of rivet hole must not be less than thickness of plate, or more than 2 × thickness of plate, the former limit to apply to thick plates and the latter to thin ones.

Diagonal pitch must not be less than 2·1 × dia. of rivet.

### *Section* 2.—*Cylindrical Furnaces.*

The thickness of plate for cylindrical furnaces is given by the formula—

$$\text{Thickness} = K\sqrt{P \times L \times D} + 0{\cdot}2 \text{ cm.}$$

Where $P$ = working pressure (kilos. per sq. cm.).
$L$ = length between stiffeners (centimetres).
$D$ = external diameter of furnace (do.).
$K$ = 0·0033 when the longitudinal joints are welded and fitted with double butt-straps, and the furnace is truly cylindrical.
$K$ = 0·0035 when the longitudinal joints are lapped only.

The thickness of plate must not, in any case, be less than that given by the formula—

$$\text{Thickness} = \frac{P \times D}{C} + 0{\cdot}2 \text{ cm.}$$

Wherein $C$ = 670 for plain furnaces without Adamson rings or other stiffeners.
$C$ = 820 for furnaces with one Adamson ring, when the unsupported distance between the ring and the ends is not more than 122 cm.
$C$ = 920 for furnaces with two Adamson rings, when the unsupported distance between the stiffeners is not more than 79 cm.
$C$ = 1160 for Fox or Morison corrugated, or Purves ribbed furnaces. In corrugated furnaces $D$ is the external diameter of the depressed corrugation, and in ribbed furnaces the external diameter of the plain portion between the ribs is to be taken.

The thickness proposed for any other type of furnace, such as Farnley, Deighton, etc., must be referred for special consideration.

For the boilers of river steamers the constant addition may be 0·1 cm. in place of 0·2 cm.

### *Section 3.—Screw and Through Stays.*

The strength of screw and through stays must not be taken higher than—

500 kilos. per sq. cm. for iron.
600 ,, ,, ,, steel.

Steel stays must not be welded.

In welded iron stays the section must be increased 20 per cent.

The pitch of screw stays in combustion chamber backs must not exceed 20 cm., and that of through stays in the steam space 45 cm.

### *Section 4.—Flat Surfaces.*

The thicknesses of stayed flat surfaces are to be those given by the formula—

$$\text{Thickness} = 0{\cdot}15 \text{ cm.} + p\sqrt{\frac{P}{K}}.$$

Wherein $p$ = greatest pitch of stays (in centimetres).
$P$ = working pressure (kilos. per sq. cm.).
$K$ = 2200 for stays screwed in and riveted over.
$K$ = 2800 for stays screwed in and fitted with nuts.

K = 3000 for stays fitted with inside and outside nuts and washers, the outer washers being three times the diameter of the stay end and half the thickness of the plate.

K = 3400 for stays fitted with inside and outside nuts and washers, the outer washers having a diameter of ·6 × pitch, and a thickness of ·66 × thickness of plate, and being riveted to the plate.

If the combustion chamber top is supported by bridge stays only, which rest on the tube-plate, and are not assisted by any sling rods from the boiler shell, the thickness of the tube-plate must not be less than that given by the formula—

$$\text{Thickness} = \frac{P \times W \times p}{1800\,(p - d)}.$$

Wherein P = working pressure (kilos. per sq. cm.).
W = width of chamber (tube-plate to back) in cm.
$p$ = pitch of tubes, in cm.
$d$ = internal diameter of plain tube, in cm.

If the tube-plates are of steel and are not exposed to the direct action of flame, the thickness may be reduced by 12½ per cent.

### *Section 5.—Combustion Chamber Bridge Stays.*

The strength of these stays is to be that given by the following formula:—

$$\text{Thickness} = \frac{P\,(L - p)\,S \times L}{K \times d^2}.$$

Wherein P = working pressure (kilos. per sq. cm.).
L = length of bridge stay, in cm.
$p$ = pitch of stay bolts in bridge stay, in cm.
S = distance apart of bridge stays, in cm.
$d$ = depth of bridge stay at centre, in cm.
K = 420 for one stay bolt in each bridge stay.
K = 630 for two or three stay bolts.
K = 720 for four stay bolts.

"Thickness" stands for the collective thickness of the two plates of the bridge stay if constructed in that way.

If the plates are of steel they may be 10 per cent. thinner; and a further reduction may be made if they are assisted by slings from above.

### *General.*

Manholes, etc., must be fitted with stiffening rings to compensate for plate cut away; they should not, as a rule, be less than 30 cm. × 40 cm., but in special cases 28 cm. × 38 cm. may be accepted. Cast-iron manhole covers cannot be accepted. It is strongly recommended that the studs of manhole covers be both screwed into the covers and riveted over on the inside.

Boilers are to be properly secured against movement both fore and aft and athwartships.

When there are several boilers they must be provided with proper shut-off valves to enable them to be worked independently.

Where the donkey boiler works at a lower pressure than the main boilers, satisfactory arrangements must be made to prevent steam passing from the latter into the former.

The space between the combustion chamber back and the back plate of the boiler should not be less than 12 cm., to allow for proper cleaning. In the cases of extra large boilers, and of boilers worked under forced draught, a greater space should be allowed.

Every marine boiler must be fitted with a blow-off valve and pipe, and also with a scum valve to draw the water from the level of the lower end of the water-gauge. The blow-off pipe must be fitted with one cock on the boiler and a second on the ship's skin.

Wood must not be used either for lagging boilers or bulkheads in the neighbourhood of boilers.

All the larger mountings are to be secured to strong flanges riveted to the shell of the boiler; and the studs securing the mountings are not to penetrate the shell.

## TESTS OF MATERIAL FOR STEEL BOILERS.

### Regulations of 1st July 1899.

All material to be of good and tough quality; any that proves brittle in working to be rejected.

A range of 5 kilos. will be allowed where ultimate tensile strength is 42 kilos. or under, and 6 kilos. where ultimate tensile is above 42 kilos.

Shell plates and other material not exposed to flame must show an elongation of at least 20 per cent. in a length of 200 millimetres; and material exposed to flame must show at least 22·5 per cent. on same length.

Material of less than 38 kilos. ultimate strength must elongate at least 25 per cent.

Bending tests are to be made on strips 25 millimetres wide, that have been heated to a dark red and quenched in water at 28° C.

They must stand bending without cracking to an angle of 180° over a radius, for shell plates, etc., of four times the thickness of the plate; and for material exposed to flame, of twice the thickness of the plate.

A tensile and a bending test is to be made from each shell plate. As regards other plates, a tensile test is to be made from every fourth plate; but, as far as possible, these test pieces should be so selected that there may be one from each furnace charge.

As regards plates other than shell plates and not exposed to flame, a bending test is to be made from each furnace charge. A bending test must be made from each plate that is exposed to flame.

Plates of over 41 kilos. tensile must be annealed if they have been worked in the fire.

Plates that are to be subjected to the direct action of flame should not have a higher ultimate tensile than 42 kilos. per square millimetre.

Steel of more than 46 kilos. ultimate tensile must not be used.

# APPENDIX D.

## LLOYD'S RULES RELATING TO USE OF ELECTRIC LIGHT ON BOARD VESSELS.

### Leads or Circuits.

1. The copper used in all wires or cables should have a conductivity of not less than that of the Engineering Standards Committee's standard, and the wires must be protected by tinning from the sulphur compounds present in the insulating material.

2. The sectional area of the copper wires in the cables should be in proportion to the current carried, and should not be less than as given in Table I., which is in accordance with the rules of the Institution of Electrical Engineers as revised in April, 1911.

3. Except for wiring fittings the sectional area of any copper conductor must not be less than that of No. 18 S.W.G. All copper conductors having a greater sectional area than No. 14 S.W.G. must be stranded.

4. The insulating material must be either vulcanized rubber of the best quality or it must be equally durable.

5. The insulation must be such that when the cables have been immersed in water for 24 hours it will, while still immersed, withstand 1000 volts for half an hour between the conductors and the water.

6. The insulation resistance should not be less than 600 megohms per statute mile at 60° F. after the cables have been immersed in water for 24 hours, the test being made after one minute's electrification at not less than 500 volts and while the cable is still immersed.

### Joints.

7. Joints in branches, or of branches with leads of small circuits, must be made in properly constructed water-tight junction boxes, or should have the copper wires thoroughly soldered and the insulation carefully carried out, all the joints being made water-tight. Conductors of larger sectional area than $^{7}/_{18}$ S.W.G. must be soldered to proper lugs for connection. Joints in flow and return wires should not be made opposite one another. All joints should be in accessible positions, none being made in bunkers, cargo spaces, or spaces which may at any time be used for carrying cargo, stores, or baggage.

8. For soldering wires, the fluxes used must not contain acid or other corrosive substances.

9. Where practicable, the leads should be placed where they can always be accessible; if they are laid in wood battens the covers should be screwed on, not nailed, and care should be taken that the

casings are so arranged that water will not lodge in them. Cables which are properly covered with protective metal sheathing, or which are protected by galvanized wire armouring, may be unencased. They should, however, be secured by screwed clips, not by staples. All sharp bends in cables should be avoided.

## Table I.

| | No. of wires and gauge in S.W.G. or in inches. | Nominal sectional area of conductors, sq. inches. | Maximum current permissible. Ampères. | No. of wires and gauge in S.W.G. or in inches. | Nominal sectional area of conductors, sq. inches. | Maximum current permissible. Ampères. |
|---|---|---|---|---|---|---|
| Fittings Wires. | 3/25 | ·0009 | 3·7 | 19/17 | ·046 | 70 |
| | 3/24 | ·0011 | 4·5 | 7/·097″ | ·050 | 74 |
| | 3/23 | ·0013 | 5·3 | 19/·058″ | ·050 | 74 |
| | 1/18 | ·0018 | 7·2 | 19/16 | ·060 | 83 |
| | 3/22 | ·0018 | 7·2 | 19/15 | ·075 | 97 |
| | 7/25 | ·0022 | 8·6 | 19/14 | ·094 | 113 |
| | 3/21 | ·0024 | 9·5 | 19/·083″ | ·100 | 118 |
| | 1/17 | ·0025 | 9·8 | 37/16 | ·117 | 130 |
| | 7/24 | ·0026 | 10·4 | 19/13 | ·125 | 134 |
| | 3/20 | ·0030 | 12·0 | 37/15 | ·150 | 152 |
| | 7/23 | ·0031 | 12·4 | 19/·101″ | ·150 | 152 |
| | 1/16 | ·0032 | 12·9 | 37/14 | ·182 | 172 |
| | 3/19 | ·0037 | 14·8 | 37/·083″ | ·200 | 184 |
| | 1/15 | ·0041 | 16·3 | 37/13 | ·250 | 214 |
| | 7/22 | ·0042 | 17·0 | 37/12 | ·300 | 240 |
| | 1/14 | ·0050 | 19 | 37/·112″ | ·350 | 264 |
| | 3/18 | ·0053 | 20 | 61/13 | ·400 | 288 |
| | 7/21 | ·0055 | 21 | 61/·097″ | ·450 | 310 |
| | 7/20 | ·0070 | 24 | 61/12 | ·500 | 332 |
| | 7/19 | ·0086 | 28 | 61/·108″ | ·550 | 357 |
| | 7/18 | ·0125 | 34 | 61/·112″ | ·600 | 384 |
| | 7/17 | ·017 | 40 | 61/·118″ | ·650 | 410 |
| | 19/20 | ·019 | 43 | 91/·098″ | ·700 | 434 |
| | 7/16 | ·022 | 46 | 91/·101″ | ·750 | 461 |
| | 19/19 | ·023 | 47 | 91/·108″ | ·800 | 488 |
| | 7/·068″ | ·025 | 50 | 91/·112″ | ·900 | 540 |
| | 7/15 | ·028 | 53 | 91/·118″ | 1·000 | 595 |
| | 19/18 | ·034 | 59 | 127/·101″ | 1·000 | 595 |
| | 7/14 | ·035 | 60 | | | |

The above sizes provide security against undesirable rise of temperature. For long leads larger wires will be required to prevent undue drop of voltage.

10. All cables which are liable to be exposed to the weather or moisture should be lead-covered, or be otherwise specially protected. Where great heat is experienced, no wood casing should be used, but the cables should be protected by iron casings, or, if they are not exposed to mechanical injury, they may be armoured with galvanized wire and fastened to decks or bulkheads with screwed clips spaced not more than 12 inches apart.

11. If cables are led through cargo spaces, coal bunkers, or spaces which may at any time be used for carrying cargo, stores, or baggage, or which are not at all times accessible, they should be strongly protected against damage, preferably by iron casings. If they are led through metal tubes, these must be strongly secured, and should be fitted so that water cannot lodge in them.

Armoured cables may be used without casings or tubes provided they are strongly secured to the underside of decks or to bulkheads by screwed clips, and provided they are armoured in conformity with the standard of the Engineering Standards Committee, as shown in Table II.

## Table II.

| Diameter of cable to be armoured measured over lead covering. | | Single wire armouring. | | Double wire armouring. | | Metal tape armouring. |
|---|---|---|---|---|---|---|
| Above. | Up to and including | Size of Galvanized Wire. | | Size of Galvanized Wire in each layer. | | Thickness of Metal Tape, two layers to be used. |
| inch. | inch. | inch. | S.W.G. | inch. | S.W.G. | inch. |
| ... | ·50 | ·072 | No. 15 | ·056 | No. 17 | ... |
| ·50 | ·75 | ·092 | No. 13 | ·072 | No. 15 | ·03 |
| ·75 | 1·00 | ·104 | No. 12 | ·080 | No. 14 | ·04 |
| 1·00 | 1·25 | ·116 | No. 11 | ·092 | No. 13 | ·04 |
| 1·25 | 1·50 | ·128 | No. 10 | ·104 | No. 12 | ·04 |
| 1·50 | 1·75 | ·144 | No. 9 | ·116 | No. 11 | ·04 |
| 1·75 | 2·00 | ·160 | No. 8 | ·128 | No. 10 | ·04 |
| 2·00 | ... | ... | | ... | | ·06 |

12. Where cables pass through beams, bulkheads, or other iron work, they should be led through special fittings of sheet lead, hard wood, or vulcanized fibre to prevent their being chafed, and where they pass through decks they should be led through metal tubes lined with wood or vulcanized fibre, and securely fastened to the decks, standing at such a height above the deck level that water cannot stand above them. Where cables pass through water-tight bulkheads the fittings must be made efficiently water-tight.

13. In vessels having spaces allotted alternately for passengers and

cargo, the lamp fittings in these spaces should be removable, and the terminals so arranged that they can be properly covered up with strong metal covers, or the whole of the fittings should be similarly provided with strong metal covers. The main switches and cut-outs should be outside these spaces, or if placed inside, they should be in strong iron boxes provided with iron covers, or otherwise securely arranged to prevent the fittings being tampered with.

## Distribution.

14. The main switchboard should be fitted if possible in the dynamo room, to which all the main circuits throughout the ship should be brought, a switch and fuse being fitted thereon for each circuit. The auxiliary switchboards for further subdivision of the current should be placed in conveniently accessible positions, and each such switchboard should be similarly fitted with a separate switch and fuse for each sub-circuit. Fuses should be fitted to each lamp circuit when these are made with reduced size of wire. If vessels are wired on the double-wire system, fuses should be fitted to each cable of these circuits.

15. In cases where electric lights are used for the mast-head light and side lights, the switches controlling these lights should be placed in a position where they can be controlled by the officer of the watch, or other responsible person, and cannot be tampered with by other members of the crew, or by passengers, &c.

16. The switchboards should be of slate or other incombustible, non-conducting, and moisture-proof material. The switches should be on the quick-break principle, and should be so constructed that they must be either full "on" or completely "off," that is, they must not be able to remain in an intermediate position. They should have ample rubbing surfaces, and their conductivity should not be less than that of the wires connected to them, and they must be incapable of forming a permanent arc when breaking circuit.

17. Fuses should be fitted to each main or auxiliary circuit on the switchboards, as near as possible to the switches of these circuits. If the switchboard is not fitted near the dynamo, or if more than one dynamo may be used on any one circuit, then fuses should also be fitted to the main cable as near as possible to each of the dynamo terminals.

18. All fuses should be fitted in easily accessible places, and as near as possible to the commencement of the cables or wires they protect. They should be mounted on slate or other incombustible bases and be arranged so that the fused metal may not be a source of danger, and where they are fitted with covers these should be incombustible.

19. All fuses should be of easily fusible and non-oxidizable metal, and should be so proportioned as to melt with a current 100 per cent. in excess of that which the cables they protect are capable of carrying, as shown in Table I. The terminals must be spaced apart or screened, so that an arc cannot be maintained when the fuse is blown. Separate

single fuses and not double pole fuses must be used on circuits where the pressure exceeds 125 volts.

20. The fuses for each cable should be made of standard dimensions, so that a large fuse cannot be used for a small cable by mistake, or, if wire fuses are used, permanent instructions should be fitted on or near each switchboard giving particulars of the proper size of fuse for each circuit.

21. In shaft passages and in damp places, all lamp switches and fuses should be of a strong water-tight pattern, or should be placed in water-tight boxes having hinged or portable water-tight covers. No switches or cut-outs are to be placed in bunkers.

22. There should be no joints in the cables leading from the dynamo to the main switchboard, nor in those leading from the main to auxiliary switchboards, nor should branches to single lamps be taken off these cables.

23. A voltmeter should be supplied with each installation. If more than one dynamo is fitted, neither being capable of the whole of the output, an ampère meter should be supplied with each dynamo, unless each dynamo is protected by extra sensitive fuses.

## Joints with Hull.

24. In vessels fitted on the single-wire system, all the joints with the hull should be placed in accessible positions. Those for single lamps or for small cables should be made with brass screws not less than three-eighths of an inch in diameter, carefully tapped into the iron or steel, having white brass washers between the wires and the vessel, or the wires should be soldered to brass-faced washers. For larger cables above $^{7}/_{18}$ S.W.G. and for the pole of dynamo the cable wires should be properly sweated into brass or copper shoes, which should be bolted to the vessel. The iron or steel where contact is made should be filed bright, and the area of contact should not be less than eight times the section of the copper of the cable.

## In Vessels carrying Petroleum.

25. The single-wire system must not be adopted for any part of the installation. Switches and fuses must not be fitted in places liable to the accumulation of petroleum vapour or gas, and all lamps in places where it is possible for gas to accumulate must, with their holders, be enclosed in air-tight fittings of thick glass. All wires in such places are to be lead-covered, or the insulation of the cables employed is to be of such a nature as not to be affected by petroleum. No joints of cables, switches, or fuses should be fitted in the pump-room, but the wires for each lamp therein should be carried to the lamp from a distributing junction box placed outside the pump-room or companion.

The following paragraphs referring to the effect of the electric light installations upon the compasses are issued as suggestions, not as Rules.

## Position of Dynamos and of Electric Motors.

26. The position and type of dynamos and electric motors should be such that the compasses will not be affected. Dynamos and large motors should be at least 30 feet from the standard compass.

## Cables.

27. In vessels fitted with continuous-current dynamos, and wired on the single-wire system, no single cable should be carried within 15 feet of any compass, and cables conveying heavy currents should be fixed at still greater distance. If it is necessary to fix any cables within this distance, then for all parts of the vessel lighted from this cable the concentric or double-wire system should be adopted, the return-wire being carried as near the flow as possible, in the vicinity of the compasses.

## Adjustment of Compasses.

28. The compasses should be adjusted with the dynamo not working, after which the vessel's head should be put upon the different courses, with the dynamo running at full speed, and on each course the indications of the compass should be noted with the dynamo running with open circuit and with all possible combinations of the current switched "on" and "off" all circuits passing near the compasses. These indications should be compared with those obtained with the dynamo stopped, and any serious deflections of the compasses remedied before the vessel sails.

# APPENDIX E.

## LLOYD'S REGISTER OF BRITISH AND FOREIGN SHIPPING.

## Refrigerating Machinery and Appliances.

On the application of the owners of vessels fitted for carrying refrigerated cargoes, the Committee will authorize their Surveyors to survey the refrigerating machinery and appliances, and in those cases where the following conditions are complied with and a satisfactory report is received from the Surveyor, certificates of these Surveys will be issued, and the notation R.M.C. (*in red*) (*i.e.* Refrigerating Machinery Certificate) will be made against the vessel's name in the Society's Register Book, and in the special list of vessels fitted with refrigerating appliances. In cases in which the refrigerating machinery and appliances are constructed under the special survey of the Society's Surveyors and to their entire satisfaction, the notation ✠ R.M.C. (*in red*) will be made in the Register Book. The name of the maker and description and number of the refrigerating machines, whether single or duplex, and the refrigerating power of the machines will be recorded in the special list in the Register Book; as will also the number and capacity of insulated cargo chambers and the nature of the insulation and the method employed for cooling the holds.

1. The insulation must be sound and in good order and of efficient construction. The details of construction showing the amount and nature of the insulating material employed in the various parts are to be reported to the Committee.

Bilge suction and sounding pipes and ballast tank air and sounding pipes, passing through insulated spaces, should be well insulated to prevent their being frozen up. No sluice valves, scuppers or drain pipes are to be fitted which will permit drainage from spaces outside of the insulated chambers into the bilges of the insulated holds.

It is *recommended* that the woodwork of the insulation over tunnel tops be fastened with screws to facilitate the examination of this part, and that extra strong battens of American elm be fitted upon it under the hatches. Insulated removable portions are to be arranged in the bulkhead insulation, where required, to give easy access to sluice valves and bilge suction roses. The bottoms, sides, and coamings of all insulated hatches and limbers should be painted to prevent decay.

Thermometer tube flanges and covers should be arranged so that water does not run down and freeze in them when taking the temperature.

Cargo battens should be provided for the floor or deck, and the sides of the chambers previous to loading the homeward cargo. Those for the sides of the chambers should be fastened, and should be at least 1½ inches in depth and 2 inches wide, one batten being placed over each frame or ground, the others being intermediately arranged. The battens for the floor and decks should be at least 2″ × 2″.

Where the brine system of refrigerating is employed, the brine circulating pipes and tanks should not be galvanized on the inside.

In cases where internally galvanized tanks and cooling pipes have been fitted, the brine cooling and return tanks, if closed, should be provided with two ventilating pipes communicating with the atmosphere. If the tanks are not closed, the cooling room should be efficiently ventilated.

2. The refrigerating machinery is to be of approved construction and of sufficient power to maintain the necessary low temperature in the cargo chambers in tropical climates when running eighteen hours per day. For cargo capacities of above 70,000 cubic feet the machinery is to be either duplex or in duplicate.

3. A sufficient amount of spare gear is to be supplied and stowed where it is readily accessible.

No spare gear will, however, be required in cases where two complete sets of refrigerating machines are fitted, *each* being of sufficient power to maintain the necessary low temperature in the cargo chambers in tropical climates when running eighteen hours per day, provided all the working parts of these machines are interchangeable.

When two similar machines are fitted, each connected to different cargo compartments, one set of spare gear suitable for either machine will suffice.

Where one single dry air machine is fitted to each compartment, the following will be required :—

1 crank shaft with eccentric sheaves, complete, or one half shaft if the halves are interchangeable.
1 piston rod and nuts for steam and air cylinders.
1 set of piston rod and connecting rod brasses.
1 piston, complete, for each steam and air cylinder.
1 cylinder cover for each pattern used in steam and air cylinders.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
1 pair main bearing brasses, complete.
Main and cut-off valves for each steam cylinder.
Balance springs and rings for steam and air slide valves.
False valve face for each pattern fitted in steam cylinders, with screws.

1 eccentric rod for each pattern used.
1 eccentric strap for each pattern used.
1 slide valve spindle and nuts for steam and air cylinders, for each pattern used.
2 main bearing bolts.
1 set of connecting rod and piston rod bolts.
Full set of air valves and seats for air compressor.
1 set of inlet and outlet valves, and 1 set valve faces (if fitted) for air expansion cylinder with screws.
1 set of valves for air, circulating and feed pumps.
1 set of escape valve springs.
50 suction springs.
50 delivery springs.
50 buffer springs.
6 tubes and 24 ferrules for condenser.
6 tubes for cooler.
6 tubes for air drying chamber.
Assorted bolts, studs, and nuts.
1 set of lead-lined nuts for air expansion cylinder cover.
A quantity of packings and joint rings.

Where one duplex or two single dry air machines are fitted to each compartment, the following will be required:—

1 crank shaft with eccentric sheaves complete, or one half shaft if the halves are interchangeable.
1 piston rod and nuts for steam and air cylinders.
1 set of connecting rod and crosshead brasses.
1 piston for H.P. steam cylinder.
1 piston, complete, for air compressor; and 1 for air expansion cylinder.
1 set of piston springs for each steam cylinder.
1 cylinder cover for each pattern used in air compression and expansion cylinders.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
Main and cut-off slide valves and spindles with nuts complete for H.P. steam cylinder.
Balance springs and rings for steam and air slide valves.
1 H.P. steam cylinder valve and valve face with screws.
1 eccentric sheave, strap, and rod for each pattern used.
1 slide valve spindle and nuts for steam and air cylinders for each pattern used.
2 main bearing bolts.
1 set of connecting rod and piston rod bolts.
Half set of air valves and seats for air compressor.
1 inlet and 1 outlet valve, and half set of valve faces (if fitted) for air expansion cylinder, with screws.
1 set of valves for air, circulating, and feed pumps.
1 set of escape valve springs.

20 suction springs.
40 delivery springs.
40 buffer springs.
6 tubes and 24 ferrules for condenser.
6 tubes for cooler and 6 for air drying chamber.
Assorted bolts, studs, and nuts.
Half set of lead-lined nuts for air expansion cylinder cover.
A quantity of packings and joint rings.

Where one single ammonia or carbonic anhydride compression machine is fitted :—

1 crank shaft with eccentric sheaves complete, or one half shaft if the halves are interchangeable.
Piston and rods complete with nuts for each steam cylinder and gas compressor.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
1 pair main bearing brasses, complete.
1 set of connecting rod and crosshead brasses.
Main and cut-off valves for steam cylinders.
1 valve spindle for each pattern used and nuts complete.
1 eccentric strap and rod for each pattern used.
1 brine pump complete.
1 cover for each pattern used.
2 main bearing bolts.
1 set of connecting rod and piston rod bolts.
1 set compressor suction and delivery valves with springs and boxes, complete.
1 set of valves for air, circulating, feed, and brine pumps.
Crank shaft for fan engine.
1 steam piston and rod, etc., for fan engine, complete.
1 pair of connecting rod brasses for fan engines, with bolts, etc., complete.
1 set of blocks for making all leather packings used.
6 tubes and 24 ferrules for condenser.
Lengths and bends of piping of each size used, together with flanges, couplings, and screwing apparatus for effecting repairs.
1 gas regulating valve.
1 distributing and 1 collecting piece with multiple branches for coils for each pattern used. If these pieces are made of forged steel, no spare pieces are required.
Sundry valves, cocks, flanges, and fittings.
Assorted bolts, studs, and nuts.
Quantity of leather packings and joint rings.

For ammonia and carbonic anhydride compression machines, the following spare gear will be required, where one duplex or two single machines are fitted to each compartment :—

1 crank shaft, or one half shaft if the halves are interchangeable.
1 steam piston rod and nut for each pattern used.

1 piston for H.P. steam cylinder, with springs, complete.
1 set of piston rings for each steam cylinder.
1 set of piston rings for each size of compressor.
1 compressor piston rod and nuts, complete, for each pattern used.
1 air pump bucket and rod.
1 circulating pump bucket and rod.
Main and cut-off slide valves for H.P. steam cylinder.
Main and cut-off valve spindles and nuts for H.P. steam cylinder.
1 eccentric sheave, strap, and rod, for each pattern used.
1 brine pump complete.
1 cover for each end of gas compressor, except where screwed plugs are used.
2 main bearing bolts.
Half set of connecting rod and piston rod bolts.
Half set compressor suction and 1 delivery valve with springs and box, complete.
1 set of valves for air, circulating, feed, and brine pumps.
1 steam piston and rod, etc., for fan engine, complete.
1 pair of connecting rod brasses for fan engines, with bolts, etc., complete.
1 set of blocks and leather for making all leather packings used.
6 tubes and 24 ferrules for condenser.
Lengths and bends of piping of each size used, together with flanges, couplings, and screwing apparatus for effecting repairs.
1 gas regulating valve.
1 distributing and 1 collecting piece with multiple branches for coils for each pattern used. If these pieces are made of forged steel, no spare pieces are required.
Sundry valves, cocks, flanges, and fittings.
Assorted bolts, studs, and nuts.
A quantity of joint rings.

In cases where an independent circulating water pump is used, and its work *cannot* be performed by the main or auxiliary engines, a duplicate pump complete should be fitted.

In cases where an independent circulating water pump is used, and its work *can* be performed by the main or auxiliary engines, a pump bucket and rod should be carried, and half set of valves for water end.

In cases where an independent surface condenser with air, circulating, and feed pumps combined is fitted, and its work *cannot* be performed by the main engines:—

1 crank shaft with eccentric sheaves complete.
1 piston and rod complete for each pattern used.
1 eccentric strap and rod complete for each pattern used.
1 slide valve and spindle complete for each pattern used.
1 pump bucket and rod complete for each pattern used.
1 set of connecting rod and piston rod bolts and nuts.

1 set of valves for air, circulating, and feed pumps.
6 condenser tubes and 12 ferrules.

In cases where an independent surface condenser with air, circulating, and feed pumps combined is fitted, and its work *can* be performed by the main engines:—

1 pump bucket and rod complete for each pattern used.
1 set of connecting rod and piston rod bolts and nuts.
Half set of valves for air, circulating, and feed pumps.
6 condenser tubes and 12 ferrules.

## Periodical Surveys.

4. In the cases of vessels engaged on voyages of more than three months' duration, a complete examination, as detailed in par. 9, is required every voyage. If this examination is made at other than the loading port a further examination is required at the loading port. (*See* par. 11.)

5. In the cases of vessels engaged on voyages of more than two and not more than three months' duration, a complete examination as above detailed is required at each alternate voyage, but at the intermediate voyage a modified examination as described in par. 10 will be sufficient, but the survey at loading port provided for in par. 11 should be held every voyage.

6. In vessels engaged on shorter over-sea voyages, the above examinations are to be held at least every three months, alternate examinations being as provided for in pars. 9 and 10, but the survey at loading port provided for in par. 11 should be held every voyage.

In the cases of vessels engaged on voyages of only a few days' duration, the complete examinations are to be held at least every three months, alternate examinations being as provided for in pars. 9 and 10, but the examination of insulation, etc., provided for in par. 11, instead of being held every voyage need only be held at intervals of four or six weeks, as may be approved in each special case.

7. *If in any case only part of the requisite examinations is held, the certificate will be endorsed with a statement of what is required to complete the survey.*

8. The date following the record R.M.C. *in red* indicates the date of the last examination of the Refrigerating Machinery and appliances as above mentioned.

When the periodical Surveys provided for in pars. 4, 5, and 6 are not held, the record R.M.C. will be expunged.

9. The complete periodical Survey required in par. 4 will consist of the following:—

The insulation throughout the holds is to be carefully examined and tested for dryness and fulness by sounding with a hammer and by boring. The test holes are to be afterwards efficiently closed. Special attention is to be paid to the spaces under the snow boxes, trunks, and hatches where dampness may accumulate, to the sides under stringers and under decks and to the tunnel tops. All limber hatches are to be removed, the limbers cleared, and the suction pipes and roses, sluices and sounding pipes are to be examined. Hatches, air trunk-ways, and thermometor tubes with their connections and fastenings are to be examined, and where trunk-ways pass through watertight bulkheads, the watertight doors are to be examined and worked.

The trunk-ways should be as airtight as practicable and their fastenings should be secure.

The steam pipes, water pipes and connections, the crank shaft and bearings, connecting rods, steam and air cylinders, pistons, slides and valves, compressors and pistons, compressor rods and glands, surface condenser and air or gas coolers, circulating, air, feed and bilge pumps, are to be carefully examined and the condensers and coolers tested if deemed necessary.

The auxiliary machinery, where fitted, is also to be examined.

The spare gear is to be examined.

In dry air machines special attention is to be given to the condition of the air expansion cylinders, their pistons and valves. In other machines special attention is to be given to the condition of the compressors, including the pistons, rods, and glands, and to the expansion valve.

The refrigerator coils and their connections and the brine pipes and tanks, where fitted, are to be carefully examined and tested if deemed necessary.

Where the brine may escape to the bilges, the cement is to be examined.

The machinery is to be examined under working conditions, and tested on the snow box or refrigerators, the time and fall of temperature being noted.

It is *recommended* that the examination of the machinery under working conditions should be made upon the vessel's arrival at a home port, before the cargo is *fully discharged.* Where brine pipes are fitted they should be examined when under frosted conditions.

10. The examination required at alternate voyages in pars. 5 and 6 will consist of the following:—

The Insulation and Trunk-ways are to be surveyed in the same manner as is required for the complete examination detailed above. (*See* par. 9.)

Provided the machinery when tested under working conditions is found to be satisfactory, the following parts only will be required to be examined at this Survey, viz. :—

Steam valves, air pump, and circulating pump.
Crank shafts and bearings.
Air and other compressors and valves.
Expansion cylinder and valves in dry air machines.
Condenser water spaces.

Sea injection valves to be opened whenever the vessel is in dry dock.
Brine pipes in holds under frosted conditions.
The spare gear.

11. If the machinery and insulation have been surveyed and passed at a home port, the further survey required at a loading port will consist of an inspection to ascertain that the dunnage battens are in good order and that no damage has been sustained to the insulation prior to the loading of the refrigerated cargo, and also of a test of the refrigerating machinery under working conditions, the temperature in the hold being noted.

If the vessel loads at more than one port, one survey only at a loading port will be required, provided it includes the examination of all insulated spaces.

If there is no Surveyor to the Society available at the loading ports, or if there is not one obtainable from a port within a reasonable distance, this survey may be held at the port where the outward cargo is discharged. If there is no Surveyor to the Society at either of these ports, the Committee will accept the report of a survey held by a Surveyor appointed by Lloyd's Agent; or (in any case where there is no Lloyd's Agent) the report of a survey by a reliable Surveyor, if available; or (if no such Surveyor is available) a report, signed by two competent Engineers, of the vessel.

# APPENDIX F.

## STEERING GEAR.

### I. Lloyd's Rules for Steering Gear.

**Section 43. 1. Rudder Stops.**—Suitable stops for the rudder should be securely fastened to the deck in way of the tiller or quadrant tiller. Where a suitable brake is fitted to the tiller or quadrant tiller, or where the steering quadrant is geared direct on to the steam steering engine, the deck stops may be dispensed with. The stops of the steam steering engines should be fitted at a smaller angle of helm than the rudder stops, so as to prevent excessive strains consequent on a rudder being forced against its stops.

2. **Spare Tiller.**—Vessels which have not two independent steering gears are to have spare tiller and gear ready for use when required.

3. **Independent Means of Steering.**—Where combined hand and steam steering gear is adopted, and in which both gears depend upon the efficiency of a keyed quadrant or tiller, independent means of steering must be provided.

Steamers above 250 feet in length are to be fitted with two independent steering gears, one of which must be a steam or other mechanical steering gear, and it is recommended that the two controlling wheels of the mechanical gear should be placed one at the gear and the other one on the navigating bridge.

4. **Protection of Steering Gear.**—In steamers above 250 feet in length, not having full poops or awning or shelter decks, the after steering wheel and gear are to be protected by a substantially constructed iron or steel deck-house or hood.

5. **Springs** or **buffers** are to be fitted to all steam steering gears of steamers.

6. **Steering Chains or Rods.**—The diameters of steering chains or rods are to be as given in Table XXV.* for the various diameters of rudder heads and the corresponding radii of quadrant tillers. Where the radius of quadrant or length of tiller adopted differs from that given in the Table, the diameter of steering chain is to be calculated from the following formula:—

$$d = \cdot 38\sqrt{\frac{D^3}{R}},$$

where $d$ = diameter of chain in inches; D = diameter of rudder head in inches according to Table for rudder heads; R = radius of quadrant or length of tiller at the centre of the chain in inches.

---

* See "Lloyd's Rules and Regulations."

7. **Leads of Steering Chains.**—Care should be taken that the leads of the steering chains are made as direct as possible, sharp nips or bends being avoided.

8. The diameters at the centre of the chain of leading block sheaves are not to be less than sixteen times that of the steering chains, and the pins of the sheaves are not to be less than twice the diameter of the chains.

## II. Board of Trade Requirements. Sec. 76.

A spare tiller (which has been properly fitted to the rudder head), relieving tackle, &c., should, in all foreign-going and home-trade steamships, be kept near the after steering gear ready for immediate service. In large steamships where the use of hand gear is impracticable and such gear is not provided, the spare tiller should be attached to the rudder head ready for immediate use, unless the working tiller is of special design and strength, in which case a spare tiller may not be required, but full particulars should be submitted for the Board's consideration. The steering gear, including chains, should be thoroughly overhauled at every survey, and taken to pieces and thoroughly examined at least once a year. The chains and blocks that are liable to interfere with or endanger the passengers or crew should be guarded by portable, but properly secured, guards.

With the view of relieving, as far as practicable, the rudders of vessels from severe and sudden shocks, springs have in some cases been fitted to the quadrant, or to the rods or chains at each side of the vessel, and the Board think that such fittings, or other efficient means, should be adopted, more particularly in the case of new vessels.

The Surveyors should note that the steam and exhaust pipes of steering engines in all new passenger steamships should be at least of the same internal diameter respectively as the steam and exhaust connections on the cylinders. The arrangement should be such that water will not readily lodge either in the cylinders or in the steam and exhaust pipes. Right-angled bends in the pipes should be avoided as much as possible, and the pipes should be used exclusively for the steering engines. When this is not the case, full particulars and sketches should be submitted to the Board for consideration.

Attention is also directed to a description of steam steering gear in which a part of the shaft, by which the helmsman actuates the controlling valve, passes through another shaft that is liable to be thrown out of line by the reaction of the spur gearing, and, consequently, is liable to jam the inside shaft to such an extent as to deprive the helmsman of the control of the steering gear. All steam steering gears should be carefully examined, and if any be found constructed in the manner described above, their use should be discouraged, and in any case they should not be approved, unless they have been tested from midship to hard over in both directions, and found satisfactory when the vessel is running at full speed.

# APPENDIX G.

## LLOYD'S REGISTER OF BRITISH AND FOREIGN SHIPPING.

## Instructions to Surveyors regarding Tests, not Prescribed in the Society's Rules, for Steel and other Materials.

1. In cases where the material is not required to be in accordance with the Society's Rules, the Committee will approve of their Surveyors testing steel plates, sections and bars, iron and steel forgings and steel castings, to suitable specifications other than those stated in the Society's Rules, if these are mutually agreed upon by the manufacturers and purchasers. In the cases of rolled material tested under these conditions, the Brand LR must not be used, this being reserved exclusively for material conforming in all respects to the requirements of the Society's Rules.

2. The Committee will also approve of their Surveyors testing iron and steel tubes and pipes, copper and brass sheets, plates and tubes, and also similar material used in constructional work, as to which there are no tests mentioned in the Society's Rules, provided the tests actually specified are considered to be of a satisfactory character and are mutually agreed upon by manufacturers and purchasers.

In cases where no such tests have been mutually agreed upon, the following specifications for the different materials are suggested as being suitable for the mutual agreement referred to, viz. :—

### Steel Forgings for Special Purposes.

In cases where it is desired to depart from the limits of strength of material provided for in the Rules of Lloyd's Register, viz. : from 28 to 32 tons per square inch, the Rules of the British Engineering Standards Committee are suggested, viz. :—

**Tensile Strength.**—A margin of 4 tons per square inch shall be allowed between the specific maximum and minimum tensile breaking strengths.

**Elongation.**—The sum of the percentage of elongation measured on a Standard test piece and of the actual tensile strength in tons per square inch shall in no case be less than 57.

**Bend Tests.**—Cold bend tests shall be made upon test pieces having a rectangular section of 1 inch wide by $\frac{3}{4}$ inch thick. The test pieces shall be machined and the edges rounded to a radius of $\frac{1}{16}$ inch, and shall be bent over the thinner section. The tests may be made by pressure or by blows. The test pieces must withstand, without fracture, being bent through an angle of 180°, the internal radius of the bend being not greater than as specified below :—

| Maximum Specified Tensile Strength of Forging. | Internal Radius of Test Piece after Bending. |
|---|---|
| | Inch |
| 32 tons per square inch . . . . . | $\frac{1}{4}$ |
| Above 32 and up to 36 tons per square inch . . | $\frac{3}{8}$ |
| ,, 36 ,, 40 ,, ,, . . | $\frac{5}{8}$ |

## Lapwelded Iron or Steel Boiler Tubes.

Two per cent. of plain tubes and 2 per cent. of stay tubes are to be subjected to a hydraulic test of 750 lbs. per square inch.

Two per cent. of plain tubes are to be expanded cold at both ends by roller expanders to an increase of $\frac{1}{40}$th of their diameter. These tubes are then to be cut up, and two pieces, each 2 inches long, taken from each for crushing and bending tests.

A piece of each selected tube, 2 inches long, shall be placed on end, and must stand hammering down until it is reduced to $1\frac{5}{8}$ inches long without showing crack or flaw. A similar test piece from each selected tube is to be flattened till its narrowest width is two-thirds the original diameter.

In the case of stay tubes where the ends are made thicker than the body of the tube, the thickening is to be done by upsetting, not by any welding process.

All the tubes are to be inspected and gauged for both diameter and thickness.

Should any of the above tests fail, two further tubes shall be taken and the test repeated on a piece or pieces cut from each of them. If the repeat tests from both tubes are satisfactory, the batch of tubes represented shall be accepted ; but, if defects are again shown, the batch of tubes represented shall be rejected.

## Charcoal Iron Lapwelded Boiler Tubes.

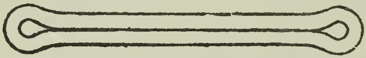
Two per cent. of plain tubes and 2 per cent. of stay tubes are to be subjected to a hydraulic test of 750 lbs. per square inch.

Two per cent. of each size of tube ordered shall be cut up to make the following tests :—

Each tube so selected must stand expanding at both ends, both hot and cold, without crack or flaw, until the diameter of the bulged end measures not less than 15 per cent. greater than the original diameter of the tube when tested hot, and 10 per cent. greater when tested cold.

A piece of each selected tube, 2 inches long, shall be placed on end, and must stand hammering down until it is reduced to $1\frac{3}{8}$ inches long, without showing crack or flaw.

A similar piece from each selected tube must stand flattening until the interior surfaces of the tube meet without showing crack or flaw, thus:



In the case of stay tubes where the ends are made thicker than the body of the tube, the thickening is to be done by upsetting, not by any welding process.

All the tubes are to be inspected and gauged for both diameter and thickness. For the thickness, a tolerance of nothing negative and plus 10 per cent. may be permitted.

Should any of the above tests fail, two further tubes shall be taken and the test repeated on a piece or pieces cut from each of them. If the repeat tests from both tubes are satisfactory, the batch of tubes represented shall be accepted; but, if defects are again shown, the batch of tubes represented shall be rejected.

## Seamless Steel Boiler Tubes.

Seamless steel boiler tubes may be finished hot or cold, as may be specified. They are to be well finished and clean inside and out, and free from laminations, surface defects, and rust, and unless otherwise specified are to be of uniform diameter throughout. Each tube is to be in a properly annealed condition when finished.

Any thickening required to the tube ends is to be done by upsetting, not by any welding process.

Two per cent. of the tubes are to be selected by the Surveyor for making the following tests:—

**Expanding Test.**—Both ends of the tubes are to withstand being expanded cold by a three-roller expander to the following increases of diameter:—

| Thickness of Tube. | Increase of Diameter, per cent. |
|---|---|
| Under $\frac{1}{8}$ inch . . . . . | $12\frac{1}{2}$ |
| $\frac{1}{8}$ to $\frac{3}{16}$ inch . . . . . | $9\frac{1}{2}$ |
| Above $\frac{3}{16}$ inch . . . . . | $6\frac{1}{2}$ |

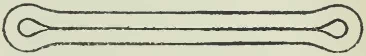
After the expanding test, the selected tubes are to be cut up and the following tests are to be made from each:—

**Tensile Tests.**—Strips about $1\frac{1}{2}$ inches wide are to be cut from the tubes and flattened. These strips may be annealed after flattening. They are to have an ultimate tensile strength of not more than 26 tons per square inch in cold finished tubes, and 27 tons in those which are hot finished, and an elongation of not less than 27 per cent. in a length of 2 inches in all cases.

**Temper Test.**—Strips about 1½ inches wide cut from the tubes and flattened, heated to a blood red, and cooled in water at about 80° F., are to stand being doubled over a radius of one-and-a-half times their thickness.

**Crushing Test.**—Pieces 2 inches long cut from the tubes are to withstand, when cold, being hammered down endwise until their length is reduced to 1 inch in the case of cold finished tubes, and to 1⅜ inches in the case of those finished hot.

**Flattening Test.**—Pieces 2 inches long, cut from cold finished tubes, are to withstand flattening by hammering, when cold, as follows, viz. :—

| | |
|---|---|
| Under ⅛ inch in thickness . . . | Till the sides are close together thus :<br> |
| ⅛ inch and up to $\frac{3}{16}$ inch . . . | Till the sides are brought to a distance apart equal to the thickness of the tube. |
| Above $\frac{3}{16}$ inch . . . . . . . . . . | Till the sides are brought to a distance apart equal to twice the thickness of the tube. |

In the case of hot finished tubes of any thickness, the flattening shall be till the sides are brought to a distance apart equal to twice the thickness of the tubes.

**Hydraulic Test.**—The whole of the tubes are to be tested by hydraulic pressure to 1000 lbs. per square inch.

Should any of the above tests fail, two further tubes shall be taken, and the test repeated on a piece or pieces cut from each of them. If the repeat tests from both tubes are satisfactory, the batch of tubes represented shall be accepted; but, if defects are again shown, the batch of tubes represented shall be rejected.

The tubes are to be gauged for thickness and for diameter.

**Tolerance as to Dimensions.**—For thickness, a tolerance of nothing negative and of plus 10 per cent. for cold finished tubes may be permitted.

For hot finished tubes the tolerance may be nothing negative and plus 15 per cent. in tubes not over 10 feet in length, and of plus 20 per cent. in longer tubes.

For diameter, in all cases a tolerance of ½ per cent. greater diameter may be permitted. In cold finished tubes a tolerance of 1 per cent. smaller diameter, in tubes not more than 2 inches diameter, and of ½ per cent. in larger tubes, may be permitted.

In hot finished tubes the tolerance of less diameter may be 2 per cent. for tubes not more than 2 inches diameter, and 1½ per cent. in larger tubes.

## Iron or Steel Steam Pipes.

These must be made of a thickness suitable for the diameter and the steam pressure at which they are to be used. If intended for use in vessels classed in Lloyd's Register, the thickness must be submitted to and approved by the Committee.

Lapwelded iron and lapwelded or seamless steel pipes are to be submitted to a hydraulic test of at least three times the intended working steam pressure.

The flanges, whether of iron or steel, are to be forged from the solid, and are to be screwed or riveted on, as may be specified. If the flanges are screwed on, the threads are to be vanishing.

In special cases the welding of flanges to the pipes may be approved, if particulars of the welding process are submitted and considered to be satisfactory.

All steam pipes should be properly annealed after the bending $\frac{\text{and}}{\text{or}}$ welding operations are completed.

## Seamless and Brazed Copper Tubes.

The tubes must be clean, smooth, and free from surface defects. At least 2 per cent. of the tubes must be cut up for test.

Tubes may be ordered either annealed or hard drawn. All test pieces shall be annealed before testing.

The tubes must stand drifting, without showing either crack or flaw, until the diameter of the drifted end measures at least 25 per cent. greater than the original diameter of the tube.

A piece of the tube shall be flattened down until the interior surfaces of the tube meet, and then the flattened tube shall be doubled over itself close, that is, bent through an angle of 180°, the bend being at right angles to the length of the tube. This test should be made both hot and cold.

All tubes shall be subjected to an internal hydraulic test to such pressure as may be specified. If no pressure is specified a suitable pressure is given by—

$$\text{Pressure in lbs. per square inch} = 4500\frac{t}{D},$$

where $t$ is the thickness and D the internal diameter in inches.

A tolerance of nothing negative and 5 per cent. positive may be allowed in the thickness.

For copper steam pipes, electro-deposited pipes must not be used. For large pipes, say 4 inches diameter and above, instead of the flattening test, the tests applicable to plates and sheets may be substituted upon strips cut longitudinally and transversely from the pipes.

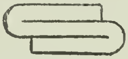
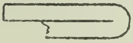
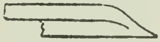
## Copper Plates and Sheets.

The plates and sheets must be clean, smooth, and free from surface defects.

Each plate or sheet must be distinctly stamped with an identification mark.

The number of tensile and bend tests to be taken are as may be specified. For large firebox plates, a tensile and a bend test should be taken from each plate. The test pieces are to be annealed before testing.

The tensile test pieces, with a width of about 1½ inches and a length of 8 inches between gauge points, should show a tensile strength of not less than 14 tons per square inch, and an elongation of at least 35 per cent.

Bend tests, both hot and cold, should be capable of being flattened over on themselves both ways thus: , without showing either crack or flaw on the outside of the bends. The cold bend test is afterwards to be drawn out cold by hammering to a feather edge, thus:  drawn to .

## Brass Condenser Plates, Condenser Tubes, &c.

Brass condenser plates are to be clean, smooth, and free from surface defects.

If tensile tests are required, the ultimate tensile strength should not be less than 22 tons per square inch, with an elongation of at least 10 per cent. measured on a Standard test piece.

Brass condenser tubes are to be seamless, of uniform diameter, and free from surface defects both inside and outside.

Each tube is to be tested by internal hydraulic pressure to 300 lbs. per square inch, the pressure being maintained while the outside of the tube is jarred along its length between suitable mallets.

The tubes are to be grouped in parcels of 100. Two of the tubes from each parcel are to be selected for the following tests :—

They are, without being previously annealed, to stand hammering on the end with a hand hammer, when held loosely, or jarring by being dropped when held horizontally from a height of 3 feet on a hard wooden floor. Pieces, 2 inches long, cut from either end of these tubes, are to be capable of being flattened till the diameter one way is 70 per cent. of the original. The tubes are also to stand heating to a dull red without fracture. They are afterwards to be cut open their whole length for an examination of the inner surfaces.

In the event of any of the above tests failing, two further tubes are to be selected from the same parcel and the failed tests are to be repeated. If these tests are satisfactory from both tubes, the parcel may be approved ; but, if any failure again occurs, the parcel of tubes represented should be rejected.

## High-Tension Brass Castings.

High-tension brass castings (including propeller blades) are to have a tensile strength of not less than 28 tons per square inch, with an elongation of not less than 15 per cent. measured on a Standard test piece.

## High-Tension Brass Rolled Rods and Forgings.

High-tension brass rolled rods and forgings are to have a tensile strength of not less than 28 tons per square inch, with an elongation of not less than 28 per cent. measured on a Standard test piece.

## Naval Brass Rods, Sheets, and Plates.

Naval brass bars are to be capable of (*a*) being hammered hot to a fine point, (*b*) being bent cold without annealing through an angle of 75° without fracture over a radius equal to the diameter of the bar.

Test pieces, at least 1½ inches wide, cut from Naval brass sheets and plates, must be capable of being bent cold through the angle specified over a radius equal to the thickness of the sample.

The tensile and bend tests for Naval brass are as follows:—

| | Ultimate Tensile Tons per square inch. | Elongation on Standard Test Piece. | Bending Angle through |
|---|---|---|---|
| | | Per cent. | ° |
| Naval brass round and hexagon bars ¾ in. and under . . | 26 | 20 | 75 |
| Naval brass round and hexagon bars above ¾ in. . . . | 22 | 20 | 75 |
| Naval brass sheets ⅜ in. thick and under, annealed . . | 26 | 30 | 180 |
| Naval brass sheets ⅜ in. thick and under, not annealed . | 26 | 25 | 135 |
| Naval brass plates above ⅜ in. up to ½ in. thick . . . | 26 | 20 | 135 |
| Naval brass plates above ½ in. up to ¾ in. thick . . . | 24 | 20 | 120 |
| Naval brass plates above ¾ in. thick . . . . . . | 22 | 22 | 90 |

# APPENDIX H.

## Lloyd's Register Rules for Diesel Engines.

### Nominal Horse Power.

The following rule is to be used for determining the nominal horse power of Diesel engines in regulating the fees for their survey, viz. :—

$$\text{N.H.P.} = \frac{N \times D^2 \sqrt{S}}{80}$$ in the case of single-acting engines of the 4-cycle type,

$$= \frac{N \times D^2 \sqrt{S}}{40}$$ in the case of single-acting engines of the 2-cycle type, and

$$= \frac{N \times D^2 \sqrt{S}}{20}$$ in the case of double-acting engines of the 2-cycle type,

where D = diameter of cylinder in inches,
S = stroke of piston in inches in ordinary reciprocating engines,
= twice the stroke of piston in the case of engines of the "Junker" type,
N = number of cylinders.

### Rules for the Construction and Survey of Diesel Engines and their Auxiliaries.

**Section 1.**—In vessels propelled by Diesel oil engines, the rules as regards machinery will be the same as those relating to steam engines, so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

### Construction.

**Section 2.**—1. In vessels built under Special Survey and fitted with Diesel engines, the engines must also be constructed under Special Survey.

2. In cases of Diesel engines being built under Special Survey, the distinguishing mark ✠ will be noted in red, thus: ✠LMC or ✠NE.

3. In order to facilitate the inspection, the plans of the machinery

are to be examined by the Surveyors, and the dimensions of the shafts are to be submitted for approval.

4. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under full power working conditions; any defects are to be pointed out as early as possible.

5. Any novelty in the construction of the machinery is to be reported to the Committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other, pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 lbs. per square inch, the diameters of the crank shaft of the main engines are not to be less than those given by the following formula:—

$$\text{Diameter of crank shaft} = \sqrt[3]{D^2 \times (AS + BL)},$$

where D = diameter of cylinder,
S = length of stroke,
L = span of bearings adjacent to a crank, measured from inner edge to inner edge.

The values of (AS + BL) are as given in the following table:—

TABLE I.

| 4-Cycle Single-Acting Engine. | 2-Cycle Single-Acting Engine. | Values of the Co-efficient. |
|---|---|---|
| 4 or 6 cyls. | 2 or 3 cyls. | ·089 S + ·056 L |
| 8 cyls. | 4 cyls. | ·099 S + ·054 L |
| 10 or 12 cyls. | 5 or 6 cyls. | ·111 S + ·052 L |
| 16 cyls. | 8 cyls. | ·131 S + ·050 L |

For auxiliary engines of the Diesel type the diameters may be 5 per cent. less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should not be less than 1·33 times and the thickness not less than 0·56 times the diameter of the shaft as found above, or, if these proportions are departed from, the webs must be of equivalent strength.

9. Where no fly-wheel is fitted, the diameter of the intermediate shaft must not be less than given by the formula:—

$$\text{Diameter of intermediate shaft} = \text{co-efficient}\ \sqrt[3]{D^2 \times S},$$

where D = diameter of cylinder,
S = stroke of piston,

and the value of the co-efficient is given by the following table:—

### TABLE II.

| 4-Cycle Single-Acting Engine. | 2-Cycle Single-Acting Engine. | Value of the Co-efficient. |
|---|---|---|
| 4 cyls. | 2 cyls. | ·456 |
| 6, 8, 10, or 12 cyls. | 3, 4, 5, or 6 cyls. | ·436 |
| 16 cyls. | 8 cyls. | ·466 |

Where the stroke is not less than 1·2 times nor more than 1·6 times the diameter of the cylinder, (·735 D + ·273 S) may be taken instead of $\sqrt[3]{D^2 \times S}$.

10. In cases where fly-wheels are fitted, the following value of the co-efficient may be taken for determining the size of the intermediate shaft abaft the fly-wheel shaft.

### TABLE III.

| 4-Cycle Single-Acting Engine. | 2-Cycle Single-Acting Engine. | Value of the Co-efficient. |
|---|---|---|
| 4 cyls. | 2 cyls. | ·405 |
| 6 ,, | 3 ,, | ·400 |
| 8 ,, | 4 ,, | ·409 |
| 10 ,, | 5 ,, | ·420 |
| 12 ,, | 6 ,, | ·427 |
| 16 ,, | 8 ,, | ·461 |

11. The diameter of the fly-wheel shaft must be at least equal to that of the crank shaft.

12. The diameter of the thrust shaft measured under the collars must be at least $\frac{21}{20}$ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

13. The diameter of the screw shaft must not be less than the diameter of the intermediate shaft (found as above) multiplied by $\left(·63 + \frac{·03\,P}{T}\right)$, but in no case must it be less than 1·07 T,

where P = the diameter of the propeller in inches,
T = the diameter of the intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in Section 18, paragraph 3, of the Rules for Engines and Boilers. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be $\frac{21}{20}$ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

14. If the designed maximum pressure in the cylinders exceeds 500 lbs. per sq. inch, the diameters of the shafting throughout must be increased in the proportion of $\sqrt[3]{\frac{\text{maximum pressure in lbs. per sq. inch}}{500}}$.

15. Where the cylinder liners are made of hard, close-grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than $\frac{1}{16}$th of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

16. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 lbs. per square inch, and must be perfectly tight at that pressure.

17. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

18. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent. above the designed maximum pressure in the cylinders and discharging where no damage can occur.

19. The air-compressors and their coolers are to be made so as to be easy of access for overhaul and adjustment.

20. In single-screw vessels, an auxiliary air-compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the manœuvring gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin-screw engines in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high-pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

## Air Receivers.

**Section 3.**—1. Compressed-air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

2. Cylindrical receivers for containing air under high pressure, used either for starting or for the injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3. **Quality of Material.**—If made of welded, or riveted, steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must have a tensile strength not exceeding 30 tons per square inch (Section 4, par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent. less than that required with boiler plates.

5. **Tensile and Bend Tests** are to be made from the material of **each** receiver. When they are welded or riveted, the tests may be made, and the thicknesses verified, before the plates are bent into cylindrical form. In the cases of seamless receivers, the thicknesses must be verified by the Surveyor before the ends are closed in, and at this time the Surveyor shall select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formulæ :—

Maximum working pressure in lbs. per square inch

$$=\frac{C \times S \times (T-2)}{D} \text{ for thicknesses of } \tfrac{5}{8} \text{ in. and above.}$$

$$=\frac{C \times S \times (T-1)}{D} \text{ for thicknesses below } \tfrac{5}{8} \text{ in.,}$$

where $S$ = minimum tensile strength of the steel material used, in tons per square inch,
$T$ = thickness of the material, in sixteenths of an inch,
$D$ = internal diameter of cylinder, in inches,
$C$ = co-efficient as per table :—

Co-efficient 77 for seamless receivers of thickness of $\frac{5}{8}$ in. and above,
,, 69 ,, ,, ,, ,, below $\frac{5}{8}$ in.
,, 54 ,, welded ,, ,, of $\frac{5}{8}$ in. and above,
,, 48 ,, ,, ,, ,, below $\frac{5}{8}$ in.

7. For flat ends welded into the cylindrical shells, the thickness must not be less than

$$T=\frac{D}{17}\sqrt{P},$$

where T = thickness, in sixteenths of an inch,
D = internal diameter, in inches,
P = working pressure, in lbs. per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates is to be tested by hydraulic pressure to twice the working pressure for pressures up to 200 lbs. per square inch. Where higher working pressures are used, the test pressure need not be more than 200 lbs. per square inch above the working pressure.

12. All receivers above 6 inches internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air-compressor is fitted with a safety valve so arranged and adjusted that no greater pressure than that permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

## Pumping Arrangements.

**Section 4.**—1. The requirements of the pumping arrangements foı the various holds, double bottoms or other ballast tanks, &c., are to be the same as required in steam vessels of the same size.

2. The engines are to be fitted with two bilge pumps, which are to be so arranged that either can be overhauled while the other is at work. In twin-screw vessels one bilge pump upon each engine will be approved. These pumps are to be arranged to draw from all compartments. Independent power-driven pumps may be fitted in lieu of these, if desired.

3. A steam pump, or equivalent power-driven pump, is also to be provided with connections to enable it to draw from all compartments

and from the sea. It must be arranged to discharge overboard and also on deck to the fire service pipes. It must have at least one suction to the engine-room bilge distinct from those connected with the bilge pumps, so that it may be used for pumping from the engine room when the bilge pumps are being used upon other parts of the vessel.

4. In addition to the above, where water ballast is used, the water-ballast pump must have one direct suction from the engine-room bilges. (This is in lieu of the bilge injection required with steam engines.)

## General.

**Section 5.**—1. For the ordinary fuel tanks the requirements of Section 49 will apply. The daily service and other separate tanks must be tested, with all their fittings, with a head of water 12 feet above their highest points. They must be fitted with air pipes discharging above the upper deck. If they are fitted with glass gauges for indicating the quantity of oil contained in them, arrangements must be made for readily shutting off the gauges in the event of the breakage of the glass, and from preventing any damage from leakage of oil.

2. Special attention must be given to the ventilation of the engine room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for cables for electric light.

4. It is recommended that all pipes conveying fuel oil should, as far as possible, be made of steel or iron, rather than copper, owing to the rapid corrosion of copper pipes when using oil containing sulphur.

## Spare Gear.

**Section 6.**—The articles mentioned in the following list will be required to be carried, viz. :—

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, &c., fitted to it.

In addition, one complete set of valves, valve seats, springs, &c., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of the cylinders of each engine.

1 piston complete, with all piston rings, studs, and nuts for the main engines.

In addition, one set of piston rings for one piston of the main and of the auxiliary Deisel engines.

1 complete set of main skew wheels for one main engine.

2 connecting rod, or piston rod top-end bolts and nuts, both for the main and the auxiliary Diesel engines.

2 connecting rod bottom-end bolts and nuts, both for the main and for the auxiliary Diesel engines.
2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.
1 set of coupling bolts for the crank shaft.
1 set of coupling bolts for the intermediate shaft.
1 complete set of piston rings for each piston of the main and of the auxiliary compressors.
1 half set of valves for the main and for the auxiliary compressors.
1 fuel pump complete for the main engine, or a complete set of all the working parts.
1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.
1 set of valves for the daily fuel supply pump.
1 set of valves for the water circulating pumps.
1 set of valves for one bilge pump.
1 set of valves for the scavenge pump, where lift valves are used.

A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.
Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

## Periodical Surveys.

**Section 7.**—1. The engines are to be submitted to survey annually, and, in addition, are to be submitted to a Special Survey upon the occasion of the vessels undergoing the Special Periodical Surveys Nos. 1, 2, and 3 prescribed in the Rules, unless the machinery has been specially surveyed within a period of twelve months, in which case the Annual Survey will be sufficient. The boilers, if fitted, are to be subjected to the same surveys as required by Section 19 of the Rules for Engines and Boilers.

2. **Special Surveys.**—At these special surveys, the main engines and the auxiliary engines are to be examined throughout, viz. :—All the cylinders, pistons, valves and valve gears, connecting rods and guides, pumps, crank, intermediate, and thrust shafts, propellers, stern bushes, sea connections and their fastenings, are to be examined. The air-compressors are also to be examined. The air-receivers are to be cleaned and examined and, if necessary, tested, as provided for in paragraph 3 of this section.

3. **Annual Surveys.**—The whole of the parts of the engines which the engineers of the vessel open up for adjustment and overhaul should be examined and reported upon. The Survey must include, for each main engine, the examination of at least 2 pistons, 2 cylinder covers and their valves, 2 connecting rods and their brasses, both top and bottom ends, 2 of the main bearings and crank shaft journals, and

1 of the tunnel bearings. If these are all satisfactory, their condition may be taken as representing that of the other similar parts.

In the auxiliary Diesel engines, a similar course must be adopted, but in this case one of each of the parts mentioned of each engine will be sufficient, if found to be satisfactory.

The valve gears of all the Diesel engines should be examined, as far as practicable, without complete dismantling.

The air receivers must be examined internally if possible, and, together with the air pipes from the compressors, must be cleaned internally by means of steam, or otherwise. If the air-receivers cannot be examined internally, they must be tested by hydraulic pressure to twice the working pressure at each alternate Annual Survey, attention being specially given to the welding of the ends and of the longitudinal joints.

## Some Ships fitted with Oil Engines—Diesel System Type.

| Name. | Date. | Gross R. Tonnage. | No. of Screws. | Engines. | | Cylinder. | | |
|---|---|---|---|---|---|---|---|---|
| | | | | N.H.P. | Cycle. | No. | Diam. | Stroke. |
| | | | | | | | ins. | ins. |
| Annam . . | 1913 | 5296 | two | 606 | four | 16 | 23·3 | 31·5 |
| Arabis . . | 1914 | 3697 | two | 302 | two | 8 | 16·5 | 33·9 |
| Ares . . . | 1914 | 3783 | two | 375 | four | 12 | 20·5 | 35·5 |
| A. von Gwinner . | 1912 | 3051 | two | 385 | two | 8 | 17·3 | 20·5 |
| Artemis . . | 1914 | 3803 | two | 385 | four | 12 | 20·5 | 35·5 |
| Bandon . . | 1909 | 3409 | one | 324 | four | 6 | 26·4 | 39·4 |
| Bayard . . | 1915 | 2900 | two | 288 | four | 12 | 19·4 | 26·0 |
| Bengkalis . . | 1915 | 1053 | one | ... | ... | 6 | 15·8 | 27·4 |
| Calgary . . | 1912 | 1639 | two | 53 | two | 8 | 11·4 | 16·9 |
| Chile . . | 1915 | 5570 | two | 568 | four | 12 | 24·8 | 37·8 |
| Fiona . . | 1914 | 5219 | two | 835 | four | 12 | 29·9 | 43·5 |
| Glengyle . . | 1915 | 6225 | two | 534 | four | 12 | 24·8 | 33·5 |
| Glenpool . . | 1913 | 5459 | two | 586 | two | 12 | 18·7 | 31·5 |
| Hamlet . . | 1915 | 5093 | two | 497 | two | 6 | 23·6 | 35·4 |
| Juno . . . | 1912 | 2345 | two | 228 | four | 6 | 22·0 | 39·4 |
| Jutlandia . . | 1912 | 4874 | two | 468 | four | 16 | 20·9 | 28·8 |
| Kangaroo . . | 1915 | 4348 | two | 398 | four | 12 | 22·0 | 29·8 |
| Pangan . . | 1909 | 3409 | one | 324 | four | 6 | 26·4 | 39·4 |
| Rolandseck . | 1912 | 1663 | one | 361 | two | 6 | 20·0 | 36·3 |
| W. E. Riedemann | 1914 | 9800 | two | 906 | two | 12 | 22·6 | 39·4 |
| Wotan . . | 1913 | 5703 | one | 550 | two | 6 | 23·4 | 43·3 |

The pumps and air-compressors must be examined and tried under working conditions. If found to be satisfactory, they need not be dismantled.

The manœuvring of the engines must be tested under working conditions.

If the examination reveals any defects, the Surveyor should recommend such further opening up as he may consider to be necessary.

4. **Record of Survey.**—If the various parts mentioned in paragraphs 2 and 3 are all found to be in a satisfactory condition and the Surveyor finds that the machinery generally is in good order, he should recommend the vessel to have a fresh record of LMC.

5. **Survey of Screw Shafts.**—Screw shafts are to be drawn and examined at intervals of not more than two years.*

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* On the application of Owners, the Committee will be prepared to give consideration to the circumstances of any special case.

# APPENDIX I.

## LLOYD'S RULES FOR SCREW SHAFTS AND STERN TUBES.

**Section 8.**—1. All shafts are to be turned all over and are to be examined when rough turned and when finished. In the case of screw shafts scrap steel is not to be used, and in no case is a mixture of scrap iron and scrap steel to be employed. It is recommended that screw shafts be made of ingot steel or forged from blooms made from rolled iron bar of good fibrous quality.

2. Gauges of an approved description for testing the truth of the crank shafts are to be supplied with all new engines, and adjusted in the presence of the Surveyor.

3. The length of the stern bush is to be at least four diameters of the shaft. It is recommended that the shaft liner should be continuous the whole length of the stern tube, and that the after end should be tapered in thickness and made watertight in the propeller boss. If the liner is made in two pieces the joint should be burned. If the liner does not fit tightly at the part between the bearings in the stern tube, the space between the shaft and the liner should be charged or "forced" with a plastic material insoluble in water and non-corrosive. If two liners are used, it is recommended that they be tapered in thickness at the ends, and that the shaft should be lapped or protected between the liners. In this case, and also if no liners are used, the diameter of the shaft should be $\frac{21}{20}$ths of that required for a shaft with a continuous liner.

4. For dimensions of shafts, see the formula.

# APPENDIX K.

## The distances to various Ports from Gibraltar and Sunderland in Nautical Miles.

| Ports. | Distance. |
|---|---|
| *From* **Gibraltar** *to*— | |
| Algiers | 410 |
| Bermuda | 2965 |
| Barcelona | 530 |
| Cadiz | 64 |
| Carthagena | 240 |
| Cette | 670 |
| Constantinople | 1780 |
| Cuxhaven | 1550 |
| Dover | 1239 |
| Finisterre | 556 |
| Genoa | 865 |
| Halifax | 2670 |
| Holyhead | 1221 |
| Leghorn | 870 |
| Liverpool | 1290 |
| London | 1326 |
| Mahon | 622 |
| Malaga | 70 |
| Malta | 980 |
| Marseilles | 690 |
| Messina | 995 |
| Naples | 970 |
| Oran | 222 |
| Palermo | 904 |
| Port Said | 1930 |
| Rome | 960 |
| Syracuse | 1020 |
| Tangiers | 30 |
| Tarragona | 490 |
| Tunis | 780 |
| Ushant | 940 |
| Valencia | 390 |
| Almeria | 150 |
| Alicante | 310 |
| Alexandria | 1805 |
| Azores | 950 |
| Boston, U.S.A. | 3080 |
| Buenos Ayres | 5410 |
| Cape Town | 5130 |
| Charleston | 3705 |
| Cape Verde Is. | 1550 |
| Cape Horn | 6500 |
| Finisterre | 556 |
| Havana | 4230 |
| Huelva | 130 |
| Jamaica | 3960 |
| Lisbon | 300 |
| Las Palmas | 850 |
| Madeira | 600 |
| New York | 3250 |
| New Orleans | 4815 |
| Newport News | 3415 |
| Oporto | 460 |
| Quebec | 3400 |
| Rio Janeiro | 4240 |
| Trinidad | 3430 |
| Panama | 4340 |
| *From* **Sunderland** *to*— | |
| Aberdeen | 150 |
| Barry | 712 |
| Belfast | 605 |
| Bristol | 742 |
| Cardiff | 720 |
| Cork | 724 |
| Dover | 267 |
| Dublin | 792 |
| Dundee | 125 |
| Falmouth | 542 |
| Glasgow (N.) | 645 |
| Holyhead (N.) | 707 |
| Hull | 123 |
| Leith | 125 |
| Limerick | 827 |
| London | 307 |
| Lynn | 145 |
| Middlesbro' | 27 |
| Newport(Mon.) | 730 |
| Plymouth | 513 |
| Sharpness | 750 |
| Southampton | 393 |
| Swansea | 700 |
| Yarmouth | 173 |
| Christiania | 536 |
| Copenhagen | 607 |
| Cronstadt | 1277 |
| Dantzig | 883 |
| Helsingfors | 1145 |
| Kiel | 695 |
| Memel | 898 |
| Malmö | 615 |
| Revel | 1112 |
| Riga | 1062 |
| Skaw | 464 |
| Stockholm | 995 |
| Archangel | 1755 |
| Amsterdam | 280 |
| Antwerp | 320 |
| Bergen | 437 |
| Boulogne | 305 |
| Bremerhaven | 366 |
| Calais | 276 |
| Dunkirk | 285 |
| Ghent | 305 |
| Gothenburg | 505 |
| Hamburg | 432 |
| Rotterdam | 295 |

*N.B.*—(S.C.) means *viâ* Suez Canal.

## The distances of various Ports from Cardiff in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Belfast | 310 | Smyrna | 2,776 | Colon | 4632 |
| Cork | 215 | Suez | 3,161 | Maranham | 3898 |
| Glasgow | 400 | Trieste | 2,806 | Monte Video | 6139 |
| Hartlepool | 705 | Venice | 2,800 | Para | 4058 |
| Hull | 638 | | | Pernambuco | 3940 |
| Liverpool | 270 | Batoum | 3,515 | Rio Janeiro | 5027 |
| London | 526 | Constantinople | 2,930 | Rio Grande do Sul | 5751 |
| Southampton | 338 | Galatz | 3,274 | | |
| Tynemouth | 728 | Odessa | 3,272 | Santos | 5227 |
| | | Taganrog | 3,519 | | |
| Copenhagen | 1108 | | | Barbadoes | 3580 |
| Cronstadt | 1776 | Aden | 4,489 | Belize | 4510 |
| Kiel | 1185 | Akyab | 7,651 | Cartagena | 4310 |
| Stettin | 1267 | Bombay | 6,154 | Demerara | 3857 |
| Stockholm | 1498 | Bushire | 6,281 | Greytown | 4657 |
| Uleaborg | 1910 | Calcutta | 7,612 | Havana | 4025 |
| | | Colombo | 6,606 | Jamaica | 4034 |
| Archangel | 2221 | Hong Kong | 9,716 | Martinique | 3555 |
| Gothenburg | 1000 | Karachi | 5,979 | New Orleans | 4559 |
| Hamburg | 821 | Madras | 6,872 | St Vincent | 3615 |
| | | Nagasaki | 10,666 | Tampico | 4846 |
| Bordeaux | 542 | Penang | 7,879 | Trinidad | 3780 |
| Bilbao | 560 | Rangoon | 7,753 | Vera Cruz | 4840 |
| Cadiz | 1093 | Shanghai | 10,466 | | |
| Lisbon | 882 | Singapore | 8,186 | Baltimore | 3285 |
| Oporto | 710 | Yokohama | 11,094 | Belle Isle | 1962 |
| Vigo | 665 | | | Boston | 2780 |
| | | Adelaide (S.C.) | 10,710 | Cape Race | 1972 |
| Algiers | 1562 | Antofagasta | 9,397 | Charleston | 3437 |
| Barcelona | 1664 | Auckland | 11,768 | Halifax | 2440 |
| Genoa | 2020 | Brisbane (S.C.) | 11,961 | New York | 3000 |
| Gibraltar | 1153 | Hobartown (S.C.) | 11,195 | Prince Edward Island | 2433 |
| Marseilles | 1844 | | | | |
| Palermo | 2057 | Mauritius (S.C.) | 9,571 | Portland | 2745 |
| Tunis | 1943 | | | Quebec | 2812 |
| | | Melbourne (S.C.) | 11,035 | | |
| Alexandria | 2943 | | | Azores | 1278 |
| Athens | 2620 | Sydney (S.C.) | 11,521 | Cape Town | 5998 |
| Beyrout | 3192 | | | Cape Verde | 2345 |
| Brindisi | 2427 | Bahia | 4,370 | Delagoa Bay | 7148 |
| Fiume | 2778 | Bermuda | 2,850 | Lagos | 4146 |
| Malta | 2133 | Buenos Ayres | 6,249 | Madeira | 1297 |
| Port Said | 3072 | Cape Horn | 7,222 | Sierra Leone | 2885 |

## The distances of various Ports from Glasgow in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| | | Gothenburg | 884 | Havana and Jamaica | 4200 |
| *From* **Glasgow** *to*— | | Hamburg (N.) | 951 | Vera Cruz | 5015 |
| Aberdeen | 520 | Bilbao | 805 | Boston, U.S.A. | 2738 |
| Barry Dock | 392 | Bordeaux | 787 | Charleston (S.C.) | 3417 |
| Belfast | 115 | Cadiz | 1,340 | | |
| Bristol | 422 | Corunna | 810 | Halifax, N.S. | 2388 |
| Cardiff | 400 | Lisbon | 1,118 | Montreal, Canada | 2907 |
| Cork | 350 | | | | |
| Deal | 696 | Barcelona | 1,908 | New York | 3120 |
| Dundee | 580 | Gibraltar | 1,401 | Portland (Me.) | 2700 |
| Falmouth | 430 | Marseilles | 2,085 | Pictou | 2476 |
| Greenock | 18 | | | Quebec | 2735 |
| Hull (N.) | 745 | | | Sable Island | 2245 |
| Limerick | 460 | Alexandria | 3,185 | St John's (N.F.) | 1907 |
| Liverpool | 220 | Dardenelles | 3,028 | St John (N.B.) | 2591 |
| London | 775 | Malta | 2,375 | | |
| Plymouth | 462 | Port Said | 3,325 | | |
| Southampton | 582 | | | Azores (S. Michael) | 1520 |
| | | Bombay | 6,396 | Bonny | 4498 |
| Copenhagen | 984 | | | Bathurst | 2728 |
| Cronstadt | 1652 | Hong Kong | 9,958 | Canaries (Las Palmas) | 1775 |
| Dantzig | 1250 | | | | |
| Kiel | 1061 | Callao | 10,382 | Cape Town | 6240 |
| Stockholm | 1380 | Sydney | 11,763 | Delagoa Bay | 7390 |
| | | Valparaiso | 9,111 | Lagos | 4388 |
| | | | | Madeira | 1540 |
| Archangel | 1981 | | | Teneriffe | 1765 |
| Bremerhaven (N.) | 901 | Buenos Ayres | 6,491 | Zanzibar (*viâ* S.C.) | 6436 |
| | | Rio Janeiro | 5,280 | | |

## The distances of various Ports from Hull in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Aberdeen | 275 | Elsinore | 610 | Iceland | 1275 |
| Barrow | 725 | Gefle | 1148 | Hammerfest | 1335 |
| Belfast | 805 | Helsingfors | 1168 | Havre | 300 |
| Barry | 630 | Heårnosand | 1248 | Ostend | 194 |
| Bristol | 660 | Kiel | 718 | Rotterdam | 212 |
| Cardiff | 638 | Königsberg | 930 | St Malo | 400 |
| Cork | 660 | Memel | 920 | Bilbao | 840 |
| Dover | 190 | Malmö | 638 | Bordeaux | 825 |
| Dublin | 710 | Norkoping | 966 | Brest | 540 |
| Dundee | 240 | Pillau | 902 | Corunna | 860 |
| Falmouth | 450 | Revel | 1135 | Cadiz | 1398 |
| Finisterre | 890 | Riga | 1085 | Huelva | 1375 |
| Glasgow (N.) | 745 | Rostock | 735 | Lisbon | 1156 |
| Hartlepool | 110 | Skaw | 490 | Oporto | 1012 |
| Harlingen | 225 | Stettin | 793 | Santander | 825 |
| Leith | 240 | Stockholm | 1019 | St Nazaire | 679 |
| Limerick | 816 | Stralsund | 718 | Vigo | 970 |
| Liverpool | 760 | Sundswall | 1199 | Villa Real | 1345 |
| Lizard | 455 | Swinemünde | 753 | | |
| London | 225 | Travemünde | 706 | Algiers | 1865 |
| Lynn | 70 | Uleaborg | 1431 | Alicante | 1755 |
| Middlesbro' | 110 | Wiborg | 1265 | Almazaron | 1681 |
| Milford Haven | 588 | | | Almeria | 1599 |
| Newport (Mon.) | 648 | Archangel | 1870 | Barcelona | 1966 |
| Plymouth | 420 | Amsterdam | 209 | Bona | 2101 |
| Southampton | 325 | Antwerp | 230 | Carthagena | 1693 |
| Sunderland | 123 | Arendal | 460 | Cette | 2118 |
| Start Point | 392 | Bergen | 500 | Genoa | 2305 |
| St Catherine's (I. Wight) | 300 | Bremerhaven | 340 | Gibraltar | 1440 |
| | | Calais | 200 | Girgenti | 2363 |
| Swansea | 618 | Christiania | 582 | Leghorn | 2310 |
| Troon (N.) | 750 | Christiansand | 423 | Malaga | 1515 |
| Tyne | 130 | Christiansund | 720 | Marseilles | 2146 |
| Ushant | 500 | Dunkirk | 200 | Messina | 2471 |
| Waterford | 626 | Fredrikshald | 548 | Naples | 2433 |
| Whitby | 90 | Fredrikstadt | 540 | Oran | 1685 |
| Yarmouth | 100 | Friederichshafen | 512 | Palermo | 2359 |
| | | | | Port Mahon | 2077 |
| Calmar | 818 | Ghent | 215 | Salerno | 2449 |
| Carlscrona | 768 | Gothenburg | 528 | Savona | 2285 |
| Copenhagen | 630 | Grimstadt | 450 | Syracuse | 2479 |
| Cronstadt | 1300 | Halmstadt | 590 | Tarragona | 1935 |
| Dantzig | 905 | Hamburg | 390 | Tangiers | 1425 |

## The distances of various Ports from Hull in Nautical Miles—*continued.*

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Tunis | 2245 | Tripoli | 2524 | Bushire | 6,566 |
| Valencia | 1841 | Venice | 3106 | Calcutta | 7,897 |
| | | Volo | 3077 | Colombo | 6,883 |
| Alexandria | 3245 | Zante | 2741 | Karachi | 6,264 |
| Alexandretta | 3505 | | | Madras | 7,157 |
| Ancona | 2996 | Batoum | 3817 | Penang | 8,164 |
| Athens | 2922 | Constantinople | 3232 | Rangoon | 8,034 |
| Bari | 2789 | Galatz | 3576 | | |
| Beyrout | 3494 | Ibrail | 3588 | Cape Town | 6,282 |
| Corfu | 2738 | Kertch | 3663 | Singapore | 8,471 |
| Corinth | 2837 | Kustenji | 3430 | Sydney | 11,806 |
| Fiume | 3080 | Odessa | 3574 | | |
| Gallipoli | 2665 | Poti | 3817 | Bahia | 4,639 |
| Jaffa | 3479 | Sebastopol | 3535 | Buenos Ayres | 6,533 |
| Larnaca | 3401 | Sulina | 3491 | Colon (Panama) | 5,000 |
| Malta | 2415 | Taganrog | 3821 | Monte Video | 6,423 |
| Patras | 2779 | Trebizonde | 3739 | Santos | 5,505 |
| Port Said | 3350 | Varna | 3381 | | |
| Ragusa | 2843 | | | Boston | 2,913 |
| Smyrna | 3078 | Aden | 4774 | New York | 3,360 |
| Trieste | 3109 | Bombay | 6438 | Quebec | 3,093 |

## The distances to various Ports from Leith in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Aberdeen | 86 | Plymouth | 620 | Bergen | 360 |
| Alloa | 28 | Southampton | 510 | Bremerhaven | 440 |
| Barry | 826 | Sunderland | 125 | Christiania | 568 |
| Cape Wrath | 260 | Tyne | 118 | Elsinore | 596 |
| Dublin (N.) | 652 | Yarmouth | 300 | Hamburg | 490 |
| Dundee | 50 | | | | |
| Dover | 374 | Copenhagen | 616 | Boston, U.S.A. | 2753 |
| Falmouth | 650 | Cronstadt | 1284 | Halifax | 2593 |
| Grangemouth | 21 | Dantzig | 875 | New York | 3180 |
| Hartlepool | 138 | Riga | 1067 | Quebec | 2933 |
| Hull | 240 | Stockholm | 1005 | St John's (N.F.) | 2105 |
| Liverpool (N.) | 686 | | | | |
| London | 415 | Antwerp | 430 | | |
| Middlesbro' | 148 | Archangel | 1691 | Iceland | 1050 |

## The distances of various Ports from Hamburg in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Aberdeen | 480 | Alexandria | 3,415 | Odessa | 3,745 |
| Barrow | 973 | Aden | 4,950 | | |
| Belfast | 900 | Bombay | 6,620 | Acapulco | 12,580 |
| Barry | 813 | Beyrout | 3,665 | Adelaide (S.C.) | 11,170 |
| Bristol | 843 | Cadiz | 1,585 | Auckland | 12,250 |
| Cardiff | 821 | Corunna | 1,035 | Callao | 10,615 |
| Cork (Queenstown) | 811 | Constantinople | 3,405 | Sydney,N.S.W. | 11,980 |
| | | Colombo | 7,070 | Valparaiso | 9,345 |
| Cuxhaven | 56 | Calcutta | 8,074 | | |
| Dover | 392 | Genoa | 2,495 | Buenos Ayres | 6,710 |
| Dublin | 881 | Gibraltar | 1,630 | Colon(Panama) | 5,085 |
| Dundee | 495 | Greytown | 5,150 | Monte Video | 6,603 |
| Falmouth | 631 | Havana | 4,530 | Pernambuco | 4,435 |
| Finisterre | 1100 | Hong Kong | 10,185 | Rio Janeiro | 5,500 |
| Glasgow | 951 | Jamaica | 4,540 | Santos | 5,685 |
| Grimsby | 376 | Lisbon | 1,350 | | |
| Hartlepool | 429 | Malta | 2,606 | Baltimore | 3,800 |
| Havre | 520 | Marseilles | 2,320 | Boston | 3,290 |
| Hull | 390 | Naples | 2,605 | Charleston | 3,970 |
| Leith | 490 | New Orleans | 5,060 | Halifax, N.S. | 2,950 |
| Limerick | 1001 | Point de Galle | 7,110 | New York | 3,510 |
| Liverpool | 964 | Port Said | 3,550 | Quebec | 3,310 |
| London | 434 | Singapore | 8,650 | Sable Island | 2,825 |
| Manchester | 1006 | Tampico | 5,350 | | |
| Milford Haven | 781 | Trinidad | 4,280 | Cape Town | 6,460 |
| Newport (Mon.) | 833 | Venice | 3,276 | Cape Verde Island | 2,795 |
| | | Vera Cruz | 5,340 | | |
| Plymouth | 601 | | | Delagoa Bay | 7,610 |
| Southampton | 481 | Batoum | 3,990 | Las Palmas | 2,000 |
| Sunderland | 432 | Galatz | 3,747 | Zanzibar (S.C.) | 6,660 |

## The distances of various Ports from Liverpool in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Belfast | 150 | Dover | 591 | Hull | 760 |
| Bristol | 292 | Dublin | 126 | Leith | 686 |
| Cardiff | 270 | Falmouth | 320 | Limerick | 470 |
| Cork | 265 | Finisterre | 739 | Lizard | 323 |
| Douglas (I.M.) | 70 | Galway | 450 | London | 680 |
| Drogheda | 120 | Glasgow | 220 | Londonderry | 217 |

## The distances of various Ports from Liverpool in Nautical Miles—*continued.*

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Milford Haven | 180 | Brindisi | 2,564 | Cape Coast Castle | 3893 |
| Newry | 130 | Fiume | 2,915 | Cape Town | 6135 |
| Plymouth | 352 | Patras | 2,608 | Cape Verde | 2470 |
| Southampton | 471 | Port Said | 3,208 | Delagoa Bay | 7283 |
| St Catherine's (I. Wight) | 478 | Smyrna | 2,913 | Lagos | 4283 |
| Swansea | 235 | Trieste | 2,944 | Las Palmas | 1665 |
| Start Point | 386 | Batoum | 3,652 | Maderia | 1435 |
| Christiania | 1005 | Constantinople | 3,067 | Sierra Leone | 3022 |
| Copenhagen | 1071 | Odessa | 3,405 | St Helena | 4560 |
| Cronstadt | 1739 | Varna | 3,216 | Teneriffe | 1660 |
| Dantzig | 1338 | Aden | 4,645 | Zanzibar (S.C.) | 6330 |
| Kiel | 1144 | Calcutta | 7,749 | Bahia | 4485 |
| Stettin | 1230 | Hong Kong | 9,853 | Buenos Ayres | 6386 |
| Stockholm | 1460 | Point de Galle | 6,786 | Cape Horn | 7355 |
| Archangel | 2058 | Rangoon | 7,890 | Colon (Panama) | 4548 |
| Antwerp | 704 | Shanghai | 10,603 | Maranham | 4035 |
| Bergen | 800 | Singapore | 8,323 | Monte Video | 6276 |
| Bremerhaven | 918 | Yokohama (*v.* S.C.) | 11,231 | Para | 4195 |
| Gothenburg | 974 | Do. (*v.* P.C.) | 8,050 | Pernambuco | 4085 |
| Hamburg | 964 | Callao (*v.* P.C.) | 5,930 | Rio Janeiro | 5175 |
| Havre | 517 | Coquimbo | 9,191 | Santos | 5345 |
| Rotterdam | 714 | Honolulu (*v.* P.C.) | 9,300 | Demerara | 3920 |
| Bilbao | 700 | Iquique | 9,775 | Havana | 4075 |
| Bordeaux | 677 | San Francisco (*v.* P.C.) | 7,836 | Jamaica | 4080 |
| Brest | 420 | Valparaiso (*v.* P.C.) | 7,202 | New Orleans | 4629 |
| Cadiz | 1230 | Do. (*v.* C.H.) | 9,006 | Tampico | 4916 |
| Lisbon | 1008 | Vancouver I. | 14,400 | Vera Cruz | 4910 |
| Oporto | 850 | Wellington, N.Z. (*v.* C.H.) | 11,605 | Baltimore | 3337 |
| Ushant | 390 | Wellington, N.Z. (*v.* P.C.) | 11,100 | Boston, U.S.A. | 2803 |
| Vigo | 800 | Melbourne (S.C.) | 11,172 | Charleston | 3485 |
| Algiers | 1700 | Sydney (S.C.) | 11,658 | Halifax | 2453 |
| Almeria | 1434 | Azores | 1,415 | New York | 3050 |
| Barcelona | 1801 | Bonny | 4,393 | P. E. Island | 2464 |
| Genoa | 2155 | | | Philadelphia | 3200 |
| Gibraltar | 1290 | | | Portland (Me.) | 2780 |
| Marseilles | 1975 | | | Quebec | 2803 |
| Naples | 2266 | | | Sable Island | 2321 |
| Palermo | 2194 | | | St John's (N.F.) | 1990 |
| Venice | 2941 | | | St John (N.B.) | 2670 |
| Alexandria | 3080 | | | | |

## The distances of various Ports from London in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Aberdeen | 445 | Boulogne | 97 | Aden | 4,676 |
| Ardrossan | 730 | Bremerhaven | 390 | Akyab | 7,838 |
| Barrow | 675 | Calais | 96 | Bombay | 6,343 |
| Barry | 518 | Cherbourg | 236 | Calcutta | 7,799 |
| Belfast | 700 | Dieppe | 162 | Colombo | 6,793 |
| Bristol | 548 | Dunkirk | 114 | Hong Kong | 9,903 |
| Cardiff | 526 | Elsinore | 682 | Karachi | 6,166 |
| Cork | 540 | Ghent | 165 | Madras | 7,059 |
| Dublin | 600 | Gothenburg | 600 | Muscat | 5,894 |
| Dundee | 415 | Hamburg | 434 | Nagasaki | 10,853 |
| Falmouth | 345 | Havre | 215 | Penang | 8,066 |
| Glasgow | 775 | Ostend | 120 | Point de Galle | 6,836 |
| Greenock | 757 | Rotterdam | 183 | Port Said | 3,250 |
| Hartlepool | 300 | St Malo | 330 | Rangoon | 7,940 |
| Hull | 225 | | | Saigon | 8,993 |
| Leith | 415 | Brest | 420 | Shanghai | 10,653 |
| Limerick | 700 | Lisbon | 1041 | Singapore | 8,373 |
| Liverpool | 680 | Vigo | 855 | Suez | 3,340 |
| Lynn | 210 | | | Yokohama (*v.* S.C.) | 11,281 |
| Milford Haven | 476 | Algiers | 1750 | | |
| Middlesbro' | 295 | Gibraltar | 1326 | Do. (*v.* P.C.) | 8,100 |
| Newport(Mon.) | 536 | Marseilles | 2040 | | |
| Newry | 630 | Naples | 2308 | Acapulco | 12,308 |
| Plymouth | 325 | | | Auckland | 11,970 |
| Sligo | 800 | | | Antofagasta | 9,582 |
| Southampton | 210 | Alexandria | 3130 | Callao | 10,340 |
| Sunderland | 307 | Athens | 2807 | Guayaquil | 11,052 |
| Swansea | 506 | Beyrout | 3379 | Valparaiso | 9,071 |
| Tyne | 315 | Brindisi | 2614 | Wellington N.Z. (*v.* C.H.) | 11,670 |
| Waterford | 526 | Jaffa | 3364 | | |
| Yarmouth | 135 | Malta | 2310 | Wellington, N.Z. (*v.* P.C.) | 11,150 |
| | | Port Said | 3250 | | |
| Christiania | 700 | Venice | 2993 | | |
| Copenhagen | 702 | | | Adelaide (S.C.) | 10,897 |
| Cronstadt | 1373 | Constantinople | 3117 | Auckland ,, | 12,600 |
| Helsingfors | 1248 | | | Batavia ,, | 8,636 |
| Riga | 1153 | | | Brisbane ,, | 12,148 |
| Stockholm | 1090 | Batoum | 3702 | Hobartown ,, | 11,382 |
| | | Galatz | 3461 | Manilla ,, | 9,758 |
| Archangel | 2050 | Odessa | 3459 | Mauritius ,, | 7,086 |
| Amsterdam | 210 | Sulina | 3376 | Melbourne ,, | 11,222 |
| Antwerp | 180 | Taganrog | 3706 | Sydney ,, | 11,708 |
| Bergen | 730 | Varna | 3266 | Wellington ,, | 12,612 |

## The distances of various Ports from London in Nautical Miles—*continued.*

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Bahia | 4530 | Barbadoes | 3810 | Portland (Me.) | 3003 |
| Bermuda | 3110 | Belize | 4740 | Quebec | 3030 |
| Buenos Ayres | 6438 | Havana | 4255 | St John's (N.F.) | 2202 |
| Cape Horn | 7458 | Jamaica | 4264 | | |
| Colon (Panama) | 4700 | Martinique | 3785 | Azores | 1400 |
| Monte Video | 6328 | Tampico | 5076 | Cape Town | 6187 |
| Para | 4247 | | | Cape Verde | 2520 |
| Rio Janeiro | 5227 | Boston, U.S.A. | 3025 | Delagoa Bay | 7337 |
| Rio Grande do Sul | 5940 | Halifax | 2680 | Lagos | 4335 |
| | | New York | 3260 | Madeira | 1480 |
| Santos | 5410 | Philadelphia | 3410 | Teneriffe | 1720 |

## The distances of various Ports from New York in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Alexandria | 3,050 | Coquimbo | 8,814 | Naples | 4,220 |
| Amsterdam | 3,325 | Cronstadt | 4,505 | New Orleans | 1,700 |
| Antwerp | 3,290 | Cape Verde | 2,890 | Point de Galle (S.C.) | 8,767 |
| Azores | 2,300 | Demerara | 2,200 | | |
| Bahia | 4,060 | Galway | 2,720 | Para | 2,900 |
| Barbadoes | 1,800 | Gibraltar | 3,250 | Pensacola | 1,630 |
| Bermuda | 690 | Glasgow | 3,120 | Pernambuco | 3,680 |
| Bombay (S.C.) | 8,251 | Greytown | 2,046 | Port Said | 5,180 |
| Bordeaux | 3,180 | Havana | 1,180 | Reval | 4,340 |
| Buenos Ayres | 5,750 | Havre | 3,170 | Rio Janeiro | 4,754 |
| Calcutta (S.C.) | 9,730 | Hong Kong (S.C.) | 11,830 | Rio Grande | 5,340 |
| Caldera | 9,080 | | | Rosario | 5,980 |
| Callao (*v.* P.C.) | 3,362 | Limon (Honduras) | 1,740 | Santos | 4,954 |
| Canton (S.C.) | 11,736 | | | Shanghai (S.C.) | 12,563 |
| Canton (*v.* Cape) | 13,880 | Lisbon | 3,030 | Singapore (S.C.) | 10,300 |
| Cape Town | 6,820 | Las Palmas | 3,401 | Stockholm | 4,255 |
| Cape Horn | 7,000 | Manilla (S.C) | 11,625 | Sydney (*v.* P.C.) | 9,811 |
| Caracas (V.) | 1,900 | Madras (S.C.) | 8,981 | San Francisco (*v.* Panama) | 5,645 |
| Charleston | 628 | Maranham | 3,100 | | |
| Colon (Panama) | 1,974 | Melbourne | 12,260 | Tampico | 2,020 |
| Constantinople | 5,030 | Monte Video | 5,650 | Trinidad | 1,940 |
| Copenhagen | 3,855 | Madeira | 3,085 | Valparaiso | 8,646 |
| | | | | Do. (*v.* P.C.) | 4,633 |

## The distances of various Ports from New York in Nautical Miles—*continued.*

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Vera Cruz | 1,990 | Cape Hatteras | 340 | Philadelphia | 220 |
| Vancouver (*v.* Panama) | 6,350 | Cape Race | 1000 | Portland (Me.) | 410 |
| | | Charleston | 628 | Sandy Hook | 18 |
| Wellington (*v.* Horn) | 10,800 | Charlestown | 425 | | |
| | | Cape May (D.) | 145 | Punta Gorda | 1200 |
| Yokohama (*v.* Suez) | 13,200 | Cape Florida | 1450 | Quebec | 1315 |
| | | Galveston | 1850 | Sable Island | 685 |
| Yokohama (*v.* Panama) | 9,823 | Halifax | 580 | Savannah | 695 |
| | | Jamaica | 1480 | St Andrews (N.B.) | 495 |
| | | Key West | 1084 | | |
| Dundee | 3,150 | Martinique | 1715 | St John (N.B.) | 520 |
| Leith | 3,180 | Montreal | 1450 | StJohn's (N.F.) | 1065 |
| | | Mobile | 1660 | Sydney (C.B.) | 810 |
| Bahamas | 860 | Nantucket Shoal | 220 | Tampa | 1300 |
| Baltimore | 410 | | | Trinidad | 1940 |
| Belle Isle | 1,085 | Nassau | 960 | Wilmington (N.C.) | 495 |
| Bermuda | 690 | Newport News | 280 | | |
| Boston | 330 | Pensacola | 1630 | Washington | 415 |

## The distances of various Ports from Southampton in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Cardiff | 338 | Gothenburg | 660 | Alexandria | 3193 |
| Dundee | 510 | Hamburg | 481 | Athens | 2618 |
| Falmouth | 160 | Rotterdam | 229 | Brindisi | 2425 |
| Greenock | 564 | Bordeaux | 520 | Corfu | 2434 |
| Hartlepool | 380 | Cadiz | 1105 | Malta | 2131 |
| Hull | 325 | Corunna | 555 | Port Said | 3069 |
| Leith | 510 | Lisbon | 869 | Venice | 2802 |
| London | 210 | Vigo | 660 | | |
| Plymouth | 137 | | | Constantinople | 2928 |
| Sunderland | 393 | Algiers | 1561 | | |
| Yarmouth | 220 | Gibraltar | 1153 | Aden | 4476 |
| | | Marseilles | 1841 | Bombay | 6143 |
| Bremerhaven | 531 | Messina | 2167 | Calcutta | 7599 |
| Christiania | 725 | Naples | 2129 | Colombo | 6593 |
| Elsinore | 780 | Palermo | 2055 | Hong Kong | 9708 |

## The distances of various Ports from Southampton in Nautical Miles—*continued.*

| Ports. | Distance. | Ports | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Karachi | 5,966 | Cape Horn | 7258 | New York | 3060 |
| Madras | 6,859 | Colon (Panama) | 4610 | Philadelphia | 3210 |
| Nagasaki | 10,653 | Monte Video | 6128 | Portland | 2803 |
| Point de Galle | 6,636 | Pernambuco | 3960 | Quebec | 2870 |
| Port Said | 3,069 | Rio Janeiro | 5027 | Sable Island | 2348 |
| Shanghai | 10,453 | Santos | 5210 | St John's (N.F.) | 2002 |
| Singapore | 8,173 | | | | |
| Suez | 3,159 | | | | |
| Yokohama | 11,081 | Barbadoes | 3610 | Azores | 1290 |
| Do. (*v.* P.C.) | 7,950 | Demerara | 3880 | Las Palmas | 1523 |
| | | Havana | 4054 | | |
| Adelaide (S.C.) | 10,697 | Jamaica | 4064 | Cape Town | 5987 |
| Auckland (C.H.) | 11,770 | New Orleans | 4589 | Cape Verde | 2320 |
| | | Tampico | 4876 | Delagoa Bay | 7137 |
| Brisbane (S.C.) | 11,948 | Trinidad | 3810 | Lagos | 4135 |
| Hobartown (S.C.) | 11,182 | Vera Cruz | 4870 | Madeira | 1280 |
| Melbourne | 11,022 | | | Mauritius (S.C.) | 8590 |
| Sydney | 11,508 | Boston, U.S.A. | 2825 | | |
| | | Cape Race | 2030 | Sierra Leone | 2874 |
| Bermuda | 2,910 | Charleston | 3495 | St Helena | 4412 |
| Buenos Ayres | 6,238 | Halifax | 2480 | Zanzibar (S.C.) | 6183 |

## The distances of various Ports from Yokohama in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| Batavia | 3,163 | New York (*v.* Panama) | 9823 | London (*v.* P.C.) | 8,100 |
| Bering Straits | 2,300 | | | Do. (S.C.) | 11,281 |
| Cape Flattery | 4,163 | San Francisco | 4750 | Liverpool (*v.* P.C.) | 8,050 |
| Hakodati | 540 | Valparaiso | 9320 | | |
| Hiogo | 400 | | | Do. (S.C.) | 11,231 |
| Hong Kong | 1,620 | | | Southampton (S.C.) | 11,081 |
| Honolulu | 3,440 | Esquimalt (Vancouver) | 4220 | | |
| Kobe | 345 | | | Do. (*v.* P.C.) | 7,950 |
| Nagasaki | 681 | Victoria (Vancouver) | 4223 | | |
| Nagasaki (*v.* Inland Sea) | 735 | Vancouver (Race I.) | 4213 | Cardiff (*v.* S.C.) | 11,094 |
| New York (*v.* Suez) | 13,630 | | | Do. (*v.* P.C.) | 7,963 |

## The distances to various Ports from Hong Kong, Halifax (N.S.), and Cape Town in Nautical Miles.

| Ports. | Distance. |
|---|---|
| *From* **Hong Kong** *to*— | |
| Amoor River | 2,400 |
| Amoy | 320 |
| Bangkok | 1,450 |
| Batavia | 1,845 |
| Bombay | 3,930 |
| Buenos Ayres | 10,600 |
| Bering Straits | 4,000 |
| Calcutta | 3,094 |
| Canton | 90 |
| Cape Town | 7,000 |
| Ceylon (Galle) | 2,950 |
| Cheefoo | 1,300 |
| Foochow | 460 |
| Formosa | 500 |
| Honolulu | 4,950 |
| Haiphong | 450 |
| Hang Chow Bay | 800 |
| Kobe | 1,375 |
| Manilla | 648 |
| Melbourne | 4,940 |
| Nagasaki | 1,067 |
| New York (*v.* Panama) | 11,297 |
| Nieuchwang | 1,390 |
| Ningpo | 743 |
| Port Arthur | 1,320 |
| Peiho River | 1,380 |
| Penang | 1,804 |
| Port Darwin | 2,371 |
| River Min | 460 |
| Rio Janeiro | 10,270 |
| Rangoon | 2,580 |
| Saigon | 916 |
| Shanghai | 900 |
| Singapore | 1,530 |
| Sourabaya | 1,923 |
| Swatow | 190 |
| Sydney | 4,400 |
| San Francisco | 6,450 |
| Thursday Is. | 2,763 |
| Taku | 1,510 |
| Tientsin | 1,540 |
| Vladivostock | 1,680 |
| Valparaiso | 10,525 |
| Woosung | 808 |
| Wei-hai-wei | 1,250 |
| Yokohama | 1,620 |
| Yangtse River Mouth | 900 |
| *From* **Halifax (N.S.)** *to*— | |
| Baltimore | 885 |
| Barbadoes | 1,900 |
| Bermuda | 750 |
| Boston | 380 |
| Cape Race | 455 |
| Charleston | 1,090 |
| Demerara | 2,280 |
| Guadaloupe | 1,720 |
| Gibraltar | 2,500 |
| Havana | 1,650 |
| Jamaica | 1,800 |
| Liverpool | 2,453 |
| Montreal | 880 |
| Nassau | 1,345 |
| Newport News | 755 |
| New York | 580 |
| Philadelphia | 725 |
| Plymouth | 2,355 |
| Portland (Me.) | 300 |
| Quebec | 760 |
| Sable Island | 170 |
| Savannah | 1,140 |
| St John's (N.F.) | 520 |
| Sydney (C.B.) | 250 |
| Trinidad | 2,050 |
| Wellington (N.C.) | 885 |
| *From* **Cape Town** *to*— | |
| Cape Amber (Madagascar) | 2300 |
| Cape Coast Castlare | 2600 |
| Cape Gardafui | 3750 |
| Cape St Mary (Madagascar) | 1600 |
| Cape Verde Islands | 3860 |
| Cape St Vincent (Portugal) | 5180 |
| Colombo | 4320 |
| Congo River | 1800 |
| Cook Town | 7000 |
| Cape Horn | 3790 |
| Callao | 6661 |
| Chinde (Zambesi) | 1630 |
| Delagoa Bay | 1150 |
| Durban | 850 |
| East London | 600 |
| Point de Galle | 4300 |
| Gibraltar | 5130 |
| Hong Kong | 7000 |
| Hobartown | 6130 |
| Iquique | 6150 |
| Jamaica | 6580 |
| Karachi | 4630 |
| Lagos | 2650 |
| Loanda | 1650 |
| Madeira | 4800 |
| Madras | 4892 |
| Maulmain | 5490 |
| Mombasa | 2480 |
| Mossamedes | 1275 |
| Mozambique | 2040 |
| Melbourne | 6025 |
| Muscat (P. Gulf) | 4650 |

## The distances to various Ports from Cape Town, Cardiff, Dover, Calcutta, and Bombay in Nautical Miles.

| Ports. | Distance. | Ports. | Distance. | Ports. | Distance. |
|---|---|---|---|---|---|
| *From* **Cape Town**—*contd.* | | Lizard Point | 172 | Cape Town | 5430 |
| | | Longships | 149 | Chittagong | 350 |
| | | Lundy Island | 65 | Colombo | 1142 |
| Manilla | 6,500 | Nash Point | 20 | Hong Kong | 3094 |
| New York | 6,820 | Start Point | 239 | Karachi | 2360 |
| Perth (W.A.) | 4,900 | St Catherine's Point | 323 | Madras | 760 |
| Port Darwin | 6,255 | | | Mauritius | 3320 |
| Port Elizabeth | 450 | Ushant | 245 | Maulmain | 800 |
| Port Nolloth | 300 | | | Penang | 1300 |
| Plymouth | 5,897 | *From* **Dover** *to—* | | Point de Galle | 1210 |
| Port Said | 5,663 | | | Java | 2596 |
| Panama | 6,840 | | | Rangoon | 780 |
| Rio Janeiro | 3,270 | Calais | 21·5 | Singapore | 1664 |
| River Niger | 2,500 | | | Tamatave | 3500 |
| Rangoon | 5,330 | Antwerp | 134 | Zanzibar | 3610 |
| Sierra Leone | 3,195 | Amsterdam | 172 | | |
| Singapore | 5,560 | Boulogne | 25·5 | | |
| St Helena | 1,800 | Bremerhaven | 380 | *From* **Bombay** *to—* | |
| Sydney (N.S.W.) | 6,480 | Copenhagen | 680 | | |
| | | Dunkirk | 37 | | |
| Sydney (C.B.) | 7,540 | Flushing | 89 | Aden | 1637 |
| San Francisco | 10,140 | Hamburg | 392 | Algoa Bay | 4260 |
| Tamatave (Madagascar) | 2,450 | Ostend | 60 | Batavia | 2736 |
| | | Rotterdam | 135 | Bushire | 1423 |
| Vancouver | 5,380 | Schomven Lightship | 91 | Cape Town | 4527 |
| Walfish Bay | 800 | | | Colombo | 940 |
| Wellington | 7,350 | Skaw | 538 | Delagoa Bay | 3560 |
| Zanzibar | 2,640 | West Hinder Lightship | 46 | Point de Galle | 980 |
| | | | | Karachi | 500 |
| | | Ushant | 298 | Madras | 1180 |
| *From* **Cardiff** *to—* | | | | Mauritius | 2570 |
| | | | | Rangoon | 2060 |
| Burlings | 834 | *From* **Calcutta** *to—* | | Seychelle Is. | 1800 |
| Dover | 431 | | | Singapore | 2458 |
| Europa Point | 1,153 | Aden | 3269 | Tamatave | 2820 |
| Finisterre | 608 | Akyab | 375 | Taticorin | 780 |
| Hartland Point | 65 | Bombay | 1915 | Zanzibar | 2600 |

# APPENDIX L.

## The N.E. Coast Inst. E. and S. Guidance Specification for Triple Compound Engines of Cargo Steamers.

1. **Indicated Horse Power.**—For calculation purposes in this specification and in average sea conditions the I.H.P. is to be found as follows:—

$$\text{I.H.P.} = \frac{D^2SN}{700}$$

$D$ = Diam. L.P. cylinder in inches.
$S$ = Stroke in feet.
$N$ = Revs. per minute. Found as per section 2.

*Note.*—The divisor is adjusted for a referred mean pressure of 30 lbs. per sq. in.

2. **Revolutions.**—$N = \dfrac{32(S+4)}{S}$.

3. **Boiler Pressure.**—180 lbs. per sq. in. (gauge).

4. **Ratios of Cylinder Areas.**—Ratio for 180 lbs. pressure.

| H.P. | M.P. | L.P. |
|---|---|---|
| 1 | About 2·74 | About 7·5 |
|  | 1 | ,, 2·74 |

5. **Cuts-off at Sea Power.**—

| | | |
|---|---|---|
| About 57 per cent. | 57 per cent. | 55 per cent. |

Each reversing lever to be fitted with expansion gear for adjusting the cuts-off in engines over 800 I.H.P.

6. **Speeds of Steam.**—The mean steam speeds to be calculated as follows:—

$$\frac{\text{Area of cylinder in sq. ins.} \times \text{pistn. speed in ft. per sec.}}{\text{Area of pipe, port, or opening in sq. ins.}} = \text{Speed in ft. per sec.}$$

Table of mean steam speeds in feet per second:—

| | H.P. | M.P. | L.P. |
|---|---|---|---|
| Main steam pipe . . . | 110 | ... | ... |
| Port opening . . . | 110 | 150 | 240 |
| Steam ports . . . | 80 | 85 | 100 |
| Exhaust passage or pipe . | 60 | 65 | 75 |

**Steam Ports and Passages.**—Width of ports to be about ·8 of diameter of cylinder; and the design of L.P. exhaust ports, passages, and pipe, to be such that the absolute pressure in L.P. cylinder, as shown by the exhaust line on indicator cards, does not differ by more than 1¼ lbs. per square inch from that recorded in the steam space of the condenser at top.

7. **Maximum Load.**—The maximum load on main working parts to be taken as the product of the area of H.P. cylinder in inches and the boiler pressure in lbs. per sq. in. (gauge).

8. **Crank Shaft.**—The diameter of crank shaft in body to be a multiple of ¼ in., and not less than ⅛ in. above Lloyd's Rule, and the proportions of the remaining parts to be not less than the following:—

1. Diameter of crank pin to be equal to diameter of shaft + ½ in.
2. Diameter of crank shaft in web to be equal to diameter of shaft + ½ in.
3. Diameter of webs to be equal to diameter of crank pin by 1·85.
4. Thickness of webs to be equal to diameter of shaft by ·62.
5. Thickness of couplings to be equal to diameter of shaft by ·25.
6. Six coupling bolts to be used for shafts up to and including 15 ins. diameter. Nine coupling bolts to be used for shafts above 15 ins. diameter.

*7. Diameter of pitch circle of coupling bolts to be 1·43 diameter of crank shaft.

*8. Diameter of coupling bolts to be equal to—

$$\cdot 7 \times \sqrt{\frac{\text{diam. of shaft}^3 \text{ in inches}}{\text{number of bolts} \times \text{diam. of pitch circle in inches}}}.$$

Bolts to be parallel.

**Note.*—These two rules may be varied, provided that equivalent strength is given. Shaft to be in three interchangeable pieces.

9. **Length of Connecting Rods.**—Length of connecting rod between centres to be twice the stroke or four times the crank radius.

**Diameter of Connecting Rod.**—Diameter at centre of length to be same diameter as piston-rod body.

**Connecting-rod Top Ends.**—Connecting rods to have single top end gudgeons for all engines having H.P. cylinders of 24 ins. diameter and under.

10. **Crosshead Guides.**—Main crosshead guides to be of the single type in all sizes of engine.

**Load on Main Crosshead Guides.**—Maximum load in lbs. on crosshead guides to be taken as:—

$$\frac{\text{Area of H.P. cylinder in sq. ins.} \times \text{boiler pressure in lbs. per sq. in. (gauge)}}{4}.$$

### 11. Maximum Pressures on Principal Bearing Surfaces.

| | Lbs. per sq. in. |
|---|---|
| Main bearings | 250 |
| Crank pins | 500 |
| Crosshead gudgeons | 1000 |
| Guide shoes (ahead) | 55 |
| Guide shoes (astern) | 110 |

Diameter by length to be taken as area of bearings.
Overall length by overall breadth as area of guide shoes.

### 12. Maximum Stresses on Principal Working Parts.

| | Lbs. per sq. in. |
|---|---|
| Ingot steel piston rod at screw | 5500 |
| Piston-rod body (after deducting ¼ in. from diameter to allow for returning) | 3000 |
| Piston and connecting-rod bolts at screw | 5500 |
| Main bearing bolts at screw | 4500 |
| Main bearing keeps (if forged) | 6000 |
| Connecting rod bottom end keep (if forged) | 7500 |
| Piston rod keep (if forged) | 7500 |
| Bolts at bottom end of valve spindle, at bottom ends of eccentric rods, and at eccentric straps | 3500 |
| Bolts at top ends of eccentric rods | 3000 |
| Screw at top end of valve spindle | 2000 |
| Pump levers—for design with double plates | 4000 |
| Pump levers—for design with single plate | 3200 |

*Notes.*—The keeps are calculated as beams with distributed loads and supported ends.

Loads on valve gear bolts found as per section 13.

Load on pump levers found as per section 22.

Levers calculated as for bending stress only.

All dimensions to be to the nearest ¼ in. above the calculated size for dimensions of 2 in. and upwards, and to the nearest ⅛ in. above the calculated size for dimensions below 2 in.

**13. Valve Gear.**—The valve-gear proportions to be determined from the load on L.P. slide valve spindle calculated as follows:—

Load in lbs. $= 4(L \times B)$.
Where $L =$ Overall length of L.P. valve face in inches.
$B =$ Overall breadth of L.P valve face in inches.

**Valve Spindles.**—Diameter at gland to be not less than $= \frac{\text{diam. piston rod at gland}}{2} + \frac{1}{2}$ in.

**Eccentric Rods.**—Eccentric rods to be parallel in the body and the diameter to be not less than $= \frac{\text{diam. piston rod at gland}}{2} + \frac{1}{4}$ in.

**Reversing Shaft.**—Diameter to be not less than $=$ diameter piston rod at gland $- 1$ in.

### Maximum Pressures on Bearing Surfaces of Valve Gears.

| | Lbs. per sq. in |
|---|---|
| Link block gudgeons . . . . . . | 500 |
| Link block slippers . . . . . . | 300 |
| Eccentric rod top end pins . . . . . | 500 |
| Eccentric sheaves (ahead and astern)—cast iron on cast iron . . . . . . . | 70 |
| Eccentric sheaves (ahead and astern)—white metal on cast iron . . . . . . . | 90 |

*Notes.*—All eccentrics and straps to be duplicates.

Eccentrics to be on body of shaft and not on couplings.

All dimensions to be to the nearest $\frac{1}{4}$ in. above the size calculated.

14. **Thrust Block.**—When of horse-shoe type the pressure on thrust collars not to exceed 60 lbs. per square inch when calculated from indicated thrust, which is to be determined as follows:—

$$\text{Lbs. Indicated thrust} = \frac{\text{I.H.P.} \times 33{,}000}{\text{propeller pitch in ft.} \times \text{revs. per min.}}.$$

15. **Main Engine-driven Reciprocating Circulating Pump (Double-acting).**—The circulating pump to be proportioned to deliver an amount of condensing water equal to 40 times the feed water, with a displacement efficiency of ·8. The feed water to be taken at 15 lbs. per I.H.P. per hour.

16. **Maximum Speeds of Circulating Water.**—The speeds of circulating water are to be calculated as follows:—

$$\frac{\text{Area of bucket in sq. ins.} \times \text{bucket speed in ft. per sec.}}{\text{area of passage in sq. ins.}} = \text{speed of water.}$$

### Approximate Speeds in Feet per Second.

| | Feet per second. |
|---|---|
| Main injection . . . . . | 11 |
| Passages in pump . . . . . | 6 |
| Valve grids . . . . . . | 7 |
| Past life of valves . . . . . | 12 |
| Discharge pipe . . . . . | 9 |

17. **Main Engine-driven Reciprocating Air Pump.**—Capacity of air pump not less than one-sixteenth of the capacity of L.P. cylinder.

18. **Condenser.**—When a condenser of the ordinary type is used, the condensing surface to be $1\frac{1}{2}$ square feet per I.H.P.

The condenser to be so arranged that the vacuum in the air pump suction pipe does not differ from that in the condenser steam space at top by more than two-tenths of an inch of mercury, or 2·72 inches of water head.

The loss of temperature between the condenser steam space at top and the condensate as discharged by the air pump, and before admixture with drains or other source of heat, not to exceed 20° F.

*Note.*—The amount of surface above specified includes an allowance for the prejudicial effects upon condensing efficiency of oil and scale on the tubes, and of residual air in the condenser.

19. **Alternative Production of Vacuum by Steam Jet.**—When a steam jet is employed for the withdrawal of air, and the condenser is suitably proportioned and arranged, the circulating water to be 32 times the feed calculated as per section 15, the air pump to be one-twentieth of the capacity of L.P. cylinder, and the loss of temperature between the steam space in condenser at top and the condensate as discharged by the air pump, and before admixture with drains or other source of heat, not to exceed 10° F.

20. **Utilisation of Heat in Auxiliary Exhaust Steam.**—Provision to be made for securing the complete absorption by the feed water of the heat in the exhausts from the various auxiliaries, including the steering engine, electric-light engine, and the evaporator.

*Note.*—With the proportions already specified, and with a reasonably airtight system, a vacuum of at least 27 ins. will be maintained in the steam space at condenser top, at sea temperatures up to 70° F., and under these conditions the whole of the heat can, in ordinary circumstances, be utilised in the feed water.

21. **Main Engine-driven Feed Pumps.**—Capacity of *each* engine-driven feed pump 1/700th of capacity of L.P. cylinder.

22. **Pump Gear.**—Load on pump gear to be calculated as follows:—Load in lbs. = 25 (area of air-pump bucket + area of circulating-pump bucket) + 15 (area of both feed-pump rams + area of both bilge-pump rams). All in sq. ins.

**Maximum Pressures on Pump-Gear Bearing Surfaces.**—

| | Lbs. per sq. in. |
|---|---|
| Pump link pins | 400 |
| Engine link pins | 300 |
| Pump lever centre gudgeon bearings | 250 |

*Note.*—For cargo vessels of large power it is recommended that the circulating and feed pumps be independently driven pumps.

23. **Evaporator.**—An evaporator to be fitted capable of producing 10 tons of fresh water per 24 hours for each 1000 I.H.P. of main engines.

24. **Prevention of Delivery of Oil to Boilers.**—It is desirable that oil should be separated from the auxiliary exhaust steam before it is used to heat the feed water, but in any case the feed water to be passed through a satisfactory filter before delivery to boilers.

25. **De-aeration of Feed Water.**—With the object of preventing boiler corrosion, adequate means to be provided for the de-aeration of the feed water before it enters the boilers.

26. **Prevention of Heat-loss by Lagging.**—All surfaces radiating utilisable heat, including the feed pipes and auxiliary exhaust pipes, to be covered with an adequate thickness of non-conducting material.

## Cargo Steamers Triple Compound Engines. N.E. Coast Inst. E. and S. Standard Rule: Revolutions $= 32(S+4) \div S$. Revolutions and Piston Speeds by it.

| Stroke. | Revolutions. | Piston Speed. | Stroke. | Revolutions. | Piston Speed. |
|---|---|---|---|---|---|
| feet. | per min. | ft. per min. | feet. | per min. | ft. per min. |
| 1·50 | 117 | 352 | 3·25 | 72 | 464 |
| 1·75 | 105 | 368 | 3·50 | 69 | 480 |
| 2·00 | 96 | 384 | 3·75 | 66 | 496 |
| 2·25 | 89 | 400 | 4·00 | 64 | 512 |
| 2·50 | 83 | 416 | 4·25 | 62 | 528 |
| 2·75 | 78 | 432 | 4·50 | 60 | 544 |
| 3·00 | 75 | 448 | 4·75 | 59 | 560 |

## Diameter of L.P. Cylinders of Triple Compound Engines by Rule: $I.H.P. = D^2 \times S \times N \div 700$.

| I.H.P. | Stroke of Pistons in inches. | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 18 | 21 | 24 | 27 | 30 | 33 | 36 | 39 | 42 | 45 | 48 | 51 | 54 | 57 |
| 300 | 34·5 | 33·8 | 33·0 | 32·4 | 31·8 | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| 350 | 37·3 | 36·5 | 35·7 | 35·0 | 34·4 | .. | .. | .. | .. | .. | .. | .. | .. | .. |
| 400 | .. | 39·0 | 38·0 | 37·4 | 36·8 | 36·0 | .. | .. | .. | .. | .. | .. | .. | .. |
| 450 | .. | 41·4 | 40·5 | 39·6 | 39·0 | 38·0 | 37·5 | .. | .. | .. | .. | .. | .. | .. |
| 500 | .. | .. | 42·7 | 41·8 | 41·0 | 40·2 | 39·6 | .. | .. | .. | .. | .. | .. | .. |
| 600 | .. | .. | 46·8 | 45·8 | 45·0 | 44·0 | 43·3 | 42·6 | .. | .. | .. | .. | .. | .. |
| 700 | .. | .. | .. | 47·8 | 48·7 | 47·6 | 46·8 | 46·0 | 45·2 | .. | .. | .. | .. | .. |
| 800 | .. | .. | .. | .. | 52·0 | 50·9 | 50·0 | 49·2 | 48·4 | 47·5 | .. | .. | .. | .. |
| 900 | .. | .. | .. | .. | 55·0 | 54·0 | 53·0 | 52·1 | 51·3 | 50·4 | 49·8 | .. | .. | .. |
| 1000 | .. | .. | .. | .. | .. | 56·9 | 56·0 | 55·0 | 54·1 | 53·1 | 52·5 | .. | .. | .. |
| 1100 | .. | .. | .. | .. | .. | 59·7 | 58·7 | 57·6 | 56·7 | 55·7 | 55·1 | .. | .. | .. |
| 1200 | .. | .. | .. | .. | .. | 62·4 | 61·3 | 60·2 | 59·2 | 58·2 | 57·5 | 56·5 | .. | .. |
| 1300 | .. | .. | .. | .. | .. | 64·9 | 63·8 | 62·7 | 61·7 | 60·6 | 59·9 | 58·8 | .. | .. |
| 1400 | .. | .. | .. | .. | .. | .. | 66·3 | 65·0 | 64·0 | 63·0 | 62·1 | 61·0 | .. | .. |
| 1500 | .. | .. | .. | .. | .. | .. | 68·6 | 67·3 | 66·2 | 65·1 | 64·3 | 63·1 | 62·2 | .. |
| 1600 | .. | .. | .. | .. | .. | .. | .. | 69·5 | 68·4 | 67·2 | 66·4 | 65·2 | 64·2 | .. |
| 1700 | .. | .. | .. | . | .. | .. | .. | .. | 70·5 | 69·2 | 68·4 | 67 2 | 66·1 | .. |
| 1800 | .. | .. | .. | .. | .. | .. | .. | .. | 72·5 | 71·2 | 70·4 | 69·2 | 68·0 | 67·1 |
| 1900 | .. | .. | .. | .. | .. | .. | .. | .. | .. | 73·2 | 72·4 | 71·1 | 70·0 | 69·0 |
| 2000 | .. | .. | .. | .. | .. | .. | .. | .. | .. | 75·0 | 74·2 | 72·9 | 71·7 | 70·7 |
| 2200 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 77·8 | 76·4 | 75·3 | 74·2 |
| 2400 | .. | .. | .. | .. | .. | .. | . | .. | .. | .. | 81·3 | 79·9 | 78·6 | 77·5 |
| 2600 | .. | .. | .. | .. | . | .. | .. | .. | .. | .. | .. | 83·0 | 81·7 | 80·5 |
| 2800 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 84·9 | 83·7 |
| 3000 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 87·9 | 86·7 |
| 3500 | .. | .. | .. | .. | .. | . | .. | .. | .. | .. | .. | .. | 95·0 | 93·6 |
| 4000 | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | .. | 100 |

# APPENDIX M.

## Russian Weights and Measures with British and Metrical Equivalents.

### Linear Measure.

1 vershok = 1·75 ins. = 45·5 millimetres.
1 arshine = 16 vershoks = 28 ins. = 711 ,,
1 sazhene = 3 arshines = 7 ft. = 2·135 metres.
1 verst = 500 sazhenes = 0·67 mile = 1073 ,,
1 foot = 6·86 vershoks = 0·43 arshine.
1 yard = 1·286 arshines = 0·4286 sazhene.
1 mile = 1·509 versts = 754·29 sazhenes.

### Square Measure.

1 sq. arshine = 0·60 sq. yd. = 0·507 sq. metre.
1 sq. sazhene = 9 sq. arshines = 49 sq. ft. = 4·552 sq. metres.
1 sq. dessiatine = 2,400 sq. sazhenes = 2·7 acres = 10,928 ,,
1 sq. verst = 250,000 sq. sazhenes = 281·2 acres.
1 square yard = 1·667 square arshines.

### Cubic Measure.

1 cub. vershok = 5·36 cub. ins. = 878 cub. cms.
1 cub. arshine = 4096 cub. vershoks = 12·7 cub. ft. = 0·3595 cub. mtr.
1 cub. sazhene = 9 cub. arshines = 12·7 cub. yds. = 9·7 cub. mts.
1 cubic foot = 0·0787 cubic arshine = 322 cubic vershoks.

### Money Values (Normal).

1 copeck = 0·0213 shilling = 0·255 penny = 2·68 centimes
100 copecks = 1 gold rouble = £0·1063 (2s. 1½d.) = 2·68 francs.
£1 sterling = 9·42 roubles. 1 shilling = 47·1 copecks.
1 penny = 3·925 copecks.

### Weights.

1 dolya = 0·6856 grains.
1 zolotnik = 96 dolyas = 2·40 drams.
1 lot = 3 zolotnik = 0·4502 oz. avoir. = 0·0126 kilograms.
1 funt = 32 lots = 0·90 lb. avoir. = 0·41 ,,
1 pood = 40 funts = 36·11 ,, = 16·41 ,,
1 berkovetz = 10 poods = 3·214 cwts. = 164 ,,
1 British ton = 62 poods. 1 cwt. = 3·1 poods. 1 lb. avoir. = 35·5 lots.

## DRY MEASURE.

1 garnet = 2·88 quarts = 3·275 litres.
1 chetverik = 9 garnets = 5·77 gallons = 26·248 ,,
1 osmina = 4 chetveriks = 11·55 pecks = 104·95 ,,
1 chetvert = 2 osminas = 5·77 bushels = 209 90 ,,
1 last = 12 chetverts = 8·66 quarters = 2419 ,,
1 quarter = 1·386 chetverts. 1 bushel = 1·385 chetveriks.

## LIQUID MEASURE.

1 chaska = 0·216 pint = 0·123 litre.
1 koushka = 10 chaskas = 2·16 pints = 1·23 litres.
1 shtoff = 12·5 chaskas = 2·70 ,, = 1·538 ,,

$$1\text{ vedro}\left\{\begin{array}{l}=10\text{ koushkas}\\=8\text{ shtoffs}\end{array}\right\}=10{\cdot}82\text{ quarts}=12{\cdot}33\ ,,$$

1 bochka = 40 vedros = 108·3 gallons = 492 ,,
1 gallon = 3·704 koushkas = 2·96 shtoffs. 1 pint = 4·63 chaskas.

# APPENDIX N.

## HYDRAULIC AND STEAM TESTS OF THE ADMIRALTY AND REGISTER SOCIETIES.

### Tests and Trials.

**Admiralty Hydraulic Tests** are as follows :—

| | |
|---|---|
| Air reserve bottles, &c., belong thereto | 4000 lbs. |
| Boiler tubes and fuel heater tubes under 2 inches | 2500 ,, |
| Solid-drawn boiler tubes above 2 inches | 1500 ,, |
| Each condenser, distiller and oil-cooler tubes | 1000 ,, |
| Feed-pump chambers, pipes, valves, &c. | 3 times boiler pressure. |
| Ash ejector pump and pipes | 600 lbs. |
| Main steam pipes, valves, &c.<br>Auxiliary pipes, valves, &c. | twice the boiler pressure. |
| Boiler-room oil-feed pumps, &c. | 400 lbs. |
| Boiler in shop and on board | 1½ × W.P. |
| Cylinders, &c., of auxiliary engines | 1½ × W.P. |
| Suction passage, &c., of feed pumps | 300 lbs. |
| Low-pressure cylinders of auxiliary engines | 235 ,, |
| Pump ends, discharge valves, &c., fire and bilge pumps | 250 ,, |
| High-pressure ahead turbine, steam end and cover | 255 ,, |
| High-pressure ahead, remainder | 200 ,, |
| After completion of this turbine | 170 ,, |
| Cruising turbine and astern turbine | 170 ,, |

**Low-Pressure Turbine :—**

| | |
|---|---|
| Astern cylinders and inlet covers | 50 lbs. |
| Ahead cylinders and inlet covers | 30 ,, |
| Exhaust chamber and passages | 30 ,, |
| On completion, the whole | 30 ,, |
| Receiver pipe and valve between cruising and high-pressure ahead turbines | 255 ,, |
| Feed suction pipes | 250 ,, |
| Pump chambers of distilling, forced lubrication, engine-room oil-feed pumps and lime tanks | 100 ,, |
| All underwater valves not over 12 inches diameter | 100 ,, |
| Receiver pipes between high-pressure ahead and low-pressure turbines | 60 ,, |
| Air pumps, auxiliary exhausts, evaporator cases, bilge suctions, sluice valves, oil-fuel suctions | 60 ,, |
| Receiver pipes, high-pressure astern and low-pressure astern turbines | 50 ,, |

| | |
|---|---|
| Working sides of underwater valves over 12 inches diameter | 50 lbs. |
| Air-pump pipes and connections | 50 ,, |
| Main eduction pipes | 30 ,, |
| Steam chambers of condensers, auxiliary engine drain tanks, steam chambers and distillers | 30 ,, |
| Shaft casings before and after fitting | 30 ,, |
| Water chambers of condensers, circulating pumps, cases, valves, &c. &c. | 25 ,, |
| The feed and oil tanks | 10 ,, |
| Boiler 0·44 ultimate strength of boiler, &c., or | W.P. + 90 ,, |

**Valves and valve boxes** to be tested with valves open and with valves shut, and afterwards tested by steam.

**Steam tests** are also to be applied to all high-pressure pistons and slide valves of auxiliary machinery; all reducing valves both before and after being fitted in the ship.

**The Italian Government** require the following tests:—

| | |
|---|---|
| High-pressure cylinder and valve chests when W.P. is less than 142 lbs. to 1·5 W.P., when over (W.P. + 71) lbs. | |
| Second cylinder and valve chests to | W.P. |
| Third cylinder and valve chests to | 0·66 ,, |
| Fourth cylinder and valve chests to | 0·33 ,, |
| Steam jackets of all cylinders | 2 × W.P. in them. |
| High-pressure casing of a turbine | 1·33 × W.P. |
| Astern-going casing of a turbine | W.P. |
| Low-pressure casing of a turbine, admission end | 0·33 ,, |
| Low-pressure casing of a turbine, exhaust end | 28½ lbs. |
| Condensers when built up and all condensers for turbines | 28½ ,, |
| Cast-iron condensers for reciprocators | 21 ,, |
| Safety valves and feed pumps | 2 × W.P. |
| Air and circulating pumps | 28½ lbs. per sq. in. |

**British Corporation** require as follows:—

| | |
|---|---|
| Hydraulic tests of boilers | 1·5 × W.P. + 50 lbs. |
| Steam pipes of copper | 2 × W.P. |
| Iron or steel | 3 × W.P. |
| Feed pipes | 2·4 W.P. when copper. |
| Steel | 3·6 × W.P. |

**Lloyd's Register** require as follows:—

| | |
|---|---|
| Hydraulic tests of boilers | 1·5 × W.P. + 50 lbs. |
| Lap-welded boiler tubes, 2 per cent. to | 750 lbs. per sq. in. |
| Seamless steel boiler tubes all to | 1000 ,, ,, |

Copper tubes, test pressure $= 4500\frac{t}{D}$.

($t$ is the thickness, and D is the internal diameter in inches.)

Condenser tubes all to 300 lbs. per sq. in.

| | |
|---|---|
| Steel and iron lap-welded steam pipes | 3 × W.P. |
| Air receivers (Diesel engines) | 2 × W.P. |

**Board of Trade** require as follows :—

| | |
|---|---|
| Hydraulic tests of boilers | 1·5 × W.P. + 50 lbs. |
| High-pressure turbine cases to | 1·33 × W.P. |
| Low-pressure condenser end | 30 lbs. per sq. in. |
| Astern-going case | W.P. |

**Admiralty Steam Trials** are as follows :—

1. 30 hours' duration trial at $\frac{1}{3}$rd H.P.
2. 30 hours, the first 8 at $\frac{4}{5}$ths H.P., and 22, remainder, at $\frac{3}{5}$ths H.P.
3. Full H.P. for 8 consecutive hours, with waste of water not exceeding $3\frac{1}{2}$ tons per 1000 H.P. per 24 hours.
4. Various short trials to test handling of engines, speed going astern, both with all and half the boilers, &c. &c., the testing of auxiliaries, &c. &c.
5. The engines are then opened up for examination, &c.
6. Final trial of 24 hours, half the time at half H.P., the remainder as may be directed, all the boilers being with steam up.

**Bureau Veritas** (*vide* pp. 641–42) require the following tests :—

*Compound engine cylinders :*

| | |
|---|---|
| (*a*) High pressure to working pressure | + 85 lbs. |
| (*b*) Medium pressure, triples to | 0·8 W.P. |
| (*c*) First medium pressure, quadruples to | W.P. |
| (*d*) Second medium pressure, quadruples to | 0·5 W.P. |
| (*e*) Low pressure of all to | 43 lbs. |
| Turbines, high pressure casings, and astern-going H.P. ends | 1·25 W.P. |

Boilers to 2 W.P. ; above 142 lbs. W.P. to W.P. + 142 lbs. On board, to W.P. + 85 lbs.

Steam and feed pipes to 2 W.P. Condensers, 28·5 lbs.

**British Marine Engineering Design and Construction Committee :—**

| | |
|---|---|
| Hydraulic tests of new boilers | 1·5 × W.P. + 50 lbs. |
| Boilers which have been on service | 1·5 × W.P. |
| Boiler smoke tubes | 1000 lbs. per sq. in. |
| Stop and safety valve chests and boiler mountings generally to | 2 × W.P. |
| Copper tubes to a pressure which will produce a stress of | 7500 lbs. |
| Copper steam pipes to | 2 × W.P. |
| Copper feed pipes (delivery) | 2·5 × W.P. |
| Wrought-iron or steel steam pipes | 3 × W.P. |
| Wrought-iron or steel feed pipes | 4 × W.P. |
| Cast-iron or steel pipes generally | 3 × W.P. |

# APPENDIX O.

## MELTING POINTS OF VARIOUS METALS AS ASCERTAINED BY PROFESSOR CARNELLY, IN DEGREES CENTIGRADE.

| | | | |
|---|---|---|---|
| Silver | between 916° and 954°. | Commercial, 960° and 1034°. | Pure, 1040°. |
| Aluminium | ,, 700 ,, 850. | | |
| Antimony | ,, 425 ,, 432. | ,, 432° ,, 440°. | |
| Bismuth | ,, 256 ,, 260. | ,, 260° ,, 270·5°. | |
| Cadmium | ,, 307 ,, 326. | | |
| Cobalt | ,, 1371 ,, 1800. | | |
| Copper | ,, 950 ,, 1100. | ,, 1100° ,, 1236°. | ,, 1330° and 1398°. |
| Iron, white cast | ,, 1075 ,, 1250. | Grey cast, 1200° ,, 1600°. | Ordinary cast, 1530°. |
| Iron, wrought-bar | ,, 1500 ,, 1600. | Pure bar, 1600°. Pure iron, 1587°. | Steel, 1375°. |
| Iridium | ,, 2200 ,, 2500. | | |
| Lead | ,, 320 ,, 334. | Pure lead, 330° and 335°. | |
| Magnesium | ,, 433 ,, 500. | | |
| Manganese | ,, 1482 ,, 1900. | | |
| Nickel | ,, 1371 ,, 1600. | | |
| Platinum | ,, 1700 ,, 2534. | | |
| Tin | 230. | | |
| Zinc | between 342 and 412. | Commercial, 414° and 450°. | Boils at 940°. |

| | | | | | |
|---|---|---|---|---|---|
| Alloys | Cu 75 + Ag 25 | melts at 1114° ; | Cu 50 + Ag 50 | melts at 941° ; |
| ,, | Pb 82 + Sb 18 | ,, 260° ; | Pb 90 + Sb 10 | ,, 240° ; |
| ,, | Pb 25 + Sn 1 | ,, 558° ; | Pb 10 + Sn 1 | ,, 541° ; |
| ,, | Pb 1 + Sn 1 | ,, 370° ; | Pb 1 + Sn 5 | ,, 378° ; |
| ,, | Pb 30·5 + Zn 69·5 | ,, 190° ; | Pb 50 + Zn 50 | ,, 202° ; |
| ,, | Sn 6 + Zn 1 | ,, 204° ; | | |
| ,, Britannia metal | Sb 18 + Zn 82 | ,, 204°. | | |

# APPENDIX P.

## Lloyd's Register of Shipping.

## New Unified Rules for the Survey and Construction of Engines and Boilers of Steam Vessels.

**Section 1.**—In steam vessels, the machinery and boilers are to be inspected throughout, the boilers tested by hydraulic pressure, and the machinery tested under steam by the Society's Engineer-Surveyors, who will furnish a report to the Committee. If found satisfactory, the Committee will thereupon grant a certificate, and insert in the Register Book the notification, "L.M.C." *in red* (*i.e.* "Lloyd's Machinery Certificate"), indicating that the machinery and boilers are certified to be in good order and safe working condition.

**Section 2.**—1. In steam vessels built under Special Survey, the machinery and boilers must also be constructed under Special Survey.

2. In cases of machinery or new boilers being built under Special Survey, the distinguishing mark ✠ will be noted in red, thus: "✠L.M.C.," or "✠N.E. & B.," or "✠N.B."

3. In order to facilitate this inspection, the plans of the machinery and boilers are to be examined, and from them the working pressure fixed.

4. Any novelty in the construction of the machinery or boilers is to be reported to the Committee.

5. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under steam; any defects, etc., to be pointed out as early as possible.

6. The Surveyors may also, if desired, compare the work as it progresses with the requirements of the specification agreed upon by the parties concerned, and certify to the conditions thereof, as far as can be seen, being satisfactorily complied with.

**Section 3.**—1. All steam vessels must have at least two entirely separate means of supplying the main boilers with feed water, when working at full power.

2. If the main feed supply is from pumps worked from the main engines, then with engines above 70 N.H.P. there must be two main feed pumps, so arranged that either can be shut off and overhauled while the other is at work. Each pump is to be capable of performing the whole work required.

3. In engines of 70 N.H.P. and under, and also in all engines of

steam fishing vessels and of tugs and tenders, one main feed pump worked from the main engines will be approved, provided the auxiliary feed pump required by clause 5 is fitted.

4. The main feed pumps may be worked by independent engines, provided they are fitted with automatic regulators for controlling their speed. If only one such pump is fitted for the main feed, the auxiliary feed pump required by clause 5 should also be fitted with an automatic speed regulator.

5. A steam pump is to be provided as an auxiliary feed supply, capable of performing the whole work required of feeding the boilers at full power. This pump is to have suctions to the hotwell and to the sea. This pump may be also used for general purposes, but in this case the suction pipes to the hotwell and sea must be entirely distinct from those to the bilges.

6. Each feed pump, main or auxiliary, is to be fitted with a relief valve.

**Section 4.**—1. The engines are to be fitted with two bilge pumps so arranged that one can be overhauled whilst the other is at work.

2. In engines of 70 N.H.P. and under, and also in engines of steam fishing vessels, and of tugs and tenders, one bilge pump worked from the main engine will be sufficient, provided the steam pump required by clause 4 is fitted.

3. In lieu of bilge pumps worked from the main engines, a separate steam pump may be fitted additional to that required by clause 4.

4. A steam pump is to be provided connected to the main bilge system as required in clause 5.

5. The bilge pumps referred to in clauses 1, 2, 3, and 4 are to be arranged to pump for each compartment of the vessel except the peaks, and to deliver overboard. At least one of the steam pumps (clauses 3, 4) is to be fitted with a direct suction to the engine-room, which can be used while the other bilge pumps are being used on other parts of the vessel (see Section 39, clause 4). All bilge suction pipes are to be fitted with strum boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should be not less than double that of the cross section of the suction pipe. The mud boxes and roses in engine-room are to be placed where they are easily accessible, and to the satisfaction of the Surveyor.

6. A bilge suction to the circulating pump is to be fitted.

7. One of the steam pumps referred to in clauses 3, 4, must be fitted to draw from the sea and to deliver water on deck.

**Section 5.**—In vessels fitted with twin screw engines, the main feed pumps and the main engine bilge pumps may be fitted, one of each on each engine, provided they are so connected that either feed pump can deal with the water from both hotwells, and that either bilge pump can draw from all parts of the vessel.

**Section 6.**—1. If an independent circulating pump is not fitted, one of the steam pumps referred to in Section 4, clauses 3, 4, must be arranged to circulate water through the condenser.

2. All discharge pipes are to have discharge valves fitted on the plating of the vessel in accessible positions, and if possible these are to be above the deep load line. The discharge valves are preferably to be of non-return type.

3. No pipes are to be carried through the bunkers without being properly protected.

**Section 7.**—1. In all steam pipes provision is to be made for expansion and contraction to take place without unduly straining the pipes. Where the provision for expansion is by means of bends in the pipes it is recommended that the various lengths of pipe should be made short of the designed lengths by amounts equal to half the calculated expansion at the temperature of the steam. Where the expansion is provided for by sleeves and packed glands, these should be of brass or gun-metal, means being provided to prevent the sleeves being drawn out when under pressure.

2. All steam pipes and also the exhaust pipes of all auxiliary engines should be provided with suitable drain cocks and pipes.

3. The exhausts from the steam steering engine and from the electric light engine are to be led by separate pipes and valves to the main engine condenser, or to an auxiliary condenser, or to the atmosphere where no auxiliary condenser is fitted.

4. For the thicknesses and tests of copper, iron, or steel steam and other pipes see Section 17.

**Section 8.**—In single screw vessels fitted with steam turbine engines arrangements shall be made so that steam can be led direct to the L.P. turbine and either the H.P. or I.P. turbine can exhaust direct to the condenser. Two turbines are to be fitted with astern wheels.

**Section 9.**—In all vessels fitted with steam engines in which the lubricating oil is circulated under pressure, a spare oil pump is to be supplied with all connections ready for immediate use, and two independent means are to be arranged for circulating water through the oil cooler.

**Section 10.**—Where the engines, thrust blocks, etc., are fitted directly upon the tank tops, the holes for the holding-down bolts are to be tapped and the bolts made watertight. In these cases a cast-iron chock is to be fitted between the tank top and the engine bed plate in way of each holding-down bolt.

**Section 11.**—A steam gauge attached to the main steam pipe and one to each steam receiver, and a vacuum gauge attached to the condenser are to be fitted in convenient places where they can be seen from the engine-room platform.

**Section 12.**—Gauges of an approved description for testing the truth of the crank shafts of reciprocating engines are to be supplied with all new engines, and adjusted in the presence of the Surveyor. Where steam turbine engines are fitted, gauges for testing the truth of the rotor, pinion, and gear wheel shafts are to be supplied, and adjusted in the presence of the Surveyor.

**Section 13.**—Where feed heaters or evaporators are fitted, the parts subjected to pressure if made of mild steel are to be constructed to meet the requirements of the rules for boilers, etc. (Sections 27 to 35). Where pressure parts are made of cast iron, the portions in direct tension are to be made 3½ times the thickness which would be required if they were made of mild steel, and those parts which are subjected to cross breaking stress only are to be made of twice the thickness which would be required if they were made of mild steel. Similar parts made of brass or gun-metal should be 2¼ and 1½ times respectively the thickness required in mild steel. These parts are to be tested by hydraulic pressure to twice the working pressure coming upon them.

## Cocks, Pipes, Sea Connections, and Pumping Arrangements.

**Section 14.**—1. All sea cocks and valves required to be fitted on the plating of the vessel are to be fixed in easily accessible places above the level of the stokehold and engine-room platforms, or are to be fitted on strong iron or steel stands of a height sufficient to lift them up to the level of these platforms.

2. The bolts securing all cocks or sea connections to the plating of the vessel are to be tapped into the plating of the vessel or fitted with countersunk heads.

3. The blow-off cocks on the plating of the vessel are to be fitted with spigots passing through the plating, and a brass or gun-metal ring on the outside. The cocks are to be so constructed that the key or spanner can only be taken off when the cock is shut. (See Section 23, clause 3.)

4. Cocks and valves connecting all bilge suction pipes are to be fixed in accessible places above the stokehold and engine-room platforms. They should be marked to indicate their purpose. If bilge suction valves are placed in the shaft tunnel, they must be made workable from a position above the bulkhead deck.

5. All valve and valve seats in connection with bilge suctions are to be of brass or gun-metal. The valves should close with a right-hand motion.

6. The arrangements of pumps, bilge suction and delivery pipes are to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect.

## Rules for Shafts.

### *Material.*

**Section 15.**—1. Shafts for marine engines may be forged from ingot steel, wrought-iron bars cross piled, scrap wrought iron, or from mild steel scrap, and they may be made by hammer, press, or by rolling mill.

2. If scrap is used, it must be selected of practically uniform quality and be thoroughly cleaned. A mixture of iron and steel scrap is not permitted in any case, and steel scrap is not to be used for screw shafts, thrust shafts, or for crank shafts.

3. When ingot steel is used it must comply with the conditions of Section 39.

4. The webs of built-up crank shafts may be made of ingot steel, scrap wrought iron, or of scrap mild steel, forged, rolled, or pressed. They may also be made of steel cast to shape, with eye-holes suitable for boring to size to fit the journals and pins. If made of cast steel, the material should have a tensile strength not exceeding 32 tons per square inch, and the sum of the tensile strength in tons per square inch and the elongation per cent. measured on a standard test piece shall be not less than 50. (See Section 38, clause 6.)

5. Couplings of wrought-iron or scrap-steel shafts may be welded to the body. Those of ingot-steel shafts may be forged from the solid or formed by upsetting the shaft ends by hydraulic pressure. Couplings may also be made separately of ingot steel, wrought iron, or scrap mild steel forged, or they may be steel castings. In such couplings provision must be made for the astern pull, either by checked rings or other approved means.

6. In ingot steel shafts made by rolling and in which the couplings are formed from the shaft by upsetting, the test pieces are to be cut from the outer edges of the couplings.

7. All shafts are to be turned all over and are to be examined when rough turned and also when finished.

## Rules for determining the Sizes of Shafts.

**Section 16.**—1. In steam turbine engines the diameters of intermediate shafts are not to be less than those given by the following formula :—

$$\text{Diameter of intermediate shaft in inches} = \sqrt[3]{\frac{\text{SHP}}{\text{R}} \times \text{F}},$$

where SHP is the maximum designed horse-power to be transmitted through the shaft when working at the maximum power ;

R is the number of revolutions per minute when working at the maximum power ;

F = 64 for ocean-going and cross-channel ships ;

= 58 for vessels working in rivers and smooth waters.

2. The diameter of the wheel shaft of a geared turbine engine, where there are two pinions geared into it, opposite or nearly opposite to one another, should not be less than 1·05 times that required for the intermediate shaft. Where there is only one pinion gearing into the wheel, or where there are two set at an angle less than 120°, the diameter of the shaft at the wheel and the adjacent journal shall not be less than 1·1 times that required for the intermediate shaft. Abaft the journal the diameter may be tapered to 1·05 times that required for the intermediate shaft.

3. The diameters of intermediate shafts for reciprocating steam engines shall not be less than those given by the following formula :—

$$\text{Diameter of intermediate shaft} = \sqrt[3]{\frac{D^2 \times S \times P}{c(r+2)}},$$

where D is the diameter in inches of the low-pressure cylinder, or the equivalent diameter where two or more low-pressure cylinders are used;
S is the stroke of piston, in inches;
P is the working pressure in the boilers, in lbs. per square inch;
$r$ is the ratio of the swept volumes of the low-pressure and of the high-pressure cylinders;
$c$ is a coefficient as per the following table :—

| General Description of Engines. | Service of Ship. | |
|---|---|---|
| | Ocean-going and Cross-channel. | Rivers and Smooth Water. |
| (*a*) 2 cranks at 90°, cylinders compound | 1800 | 2000 |
| (*b*) 1 crank or 2 cranks at 180°, cylinders compound . . . . . . | 1260 | 1400 |
| (*c*) 3 cranks at 120°, cylinders compound, triple, or quadruple . | 2000 | 2250 |
| (*d*) 4 cranks balanced, cylinders compound, triple, or quadruple . | 2000 | 2250 |
| (*e*) 4 cranks at 90°, cylinders quadruple | 1900 | 2150 |

4. Crank shafts for reciprocating engines shall have a diameter not less than 1·05 times that required for the intermediate shafts.

5. The diameter between the collars of thrust shafts transmitting torque shall not be less than 1·05 times that required for the intermediate shafts. This may be tapered down outside the collars to the size required for the intermediate shafts.

6. In twin or multiple screw ships, where intermediate shafts pass through stern tubes, the diameter of the tube shafts shall in no part be less than 1·05 times that required for the intermediate shafts, and any part of them in the tube exposed, or likely to be exposed, to sea water, shall not be less than 1·075 times that required for the intermediate shafts.

7. Tail shafts (shafts carrying the screw propellers) within the stern bush, whether within the stern tube or in the bracket, shall have a diameter not less than that given by the following formula :—

$$\text{Diameter of tail shaft under the liners, in inches} = D + \frac{P}{C},$$

where D is the diameter, in inches, required for the intermediate shaft;

P is the diameter, in inches, of the propeller;

C is a coefficient as given in the following table:—

| | Values of C. | |
|---|---|---|
| | Where Shafts are fitted with Continuous Liners. | Where Shafts are not fitted with Continuous Liners. |
| For shafts for turbine engines . . | 144 | 120 |
| For shafts for reciprocating engines for ocean and cross-channel services . | 120 | 100 |
| For shafts for reciprocating engines for river and smooth-water services . | 144 | 120 |

8. Tail shafts which are in the stern tubes should have the end forward of the stern gland tapered down to 1·05 times the diameter of the intermediate shafts.

9. In fast-running machinery, in order to prevent intermediate shafts from bending to such an extent as to cause whipping, the distance between the edges of adjacent bearings should not exceed that given by the following formulæ:—

Greatest length between adjacent bearings in feet

$$=C\sqrt{\frac{D}{R}} \text{ in the cases of solid shafts,}$$

or

$$=C\sqrt[4]{\frac{D^2+d^2}{R^2}} \text{ in the cases of hollow shafts;}$$

where D is the outside diameter of the shafts, in inches;

$d$ is the diameter of the hole in the cases of hollow shafts, in inches;

R is the maximum number of revolutions of the shaft per minute;

C is 125 for outboard shafts;
145 for inboard intermediate shafts;
175 for intermediate shafts in ships engaged exclusively in smooth-water services.

**10.** In solid couplings the thickness of the flanges at the coupling-bolt holes shall not be less than the diameter required for the coupling bolts. For tail shafts the thickness of the coupling shall not be less than one-quarter of the diameter required for the intermediate shaft.

11. The radius of the fillet connecting the coupling flange to the

body of the shaft shall not be less than 0·125 times the diameter of the shaft.

12. The diameter of the coupling bolts shall not be less than given by the following formula:—

$$\left.\begin{array}{r}\text{Diameter of coupling bolts in inches,}\\ \text{measured at the face of the coupling}\end{array}\right\} = \sqrt{\frac{D^3}{3{\cdot}5n \times r}},$$

where D is the diameter required for the intermediate shaft, in inches;
*n* is the number of bolts in the coupling;
*r* is the distance of the centre of the bolt holes from the centre of the shaft, in inches.

13. The diameter over the screw-thread of the coupling-bolts should be at least $\frac{1}{8}$ inch less than that of the body of bolts which do not exceed $1\frac{3}{4}$ inches in diameter, at least $\frac{1}{4}$ inch less for bolts over $1\frac{3}{4}$ inches and up to 3 inches in diameter, and may be $\frac{1}{2}$ inch less in bolts over 3 inches in diameter. The points of the bolts should be reduced below the thread and made taper for a length of $\frac{3}{8}$ of the diameter of the bolt, so that the bolts may be driven out without damaging the thread.

14. In built crank shafts the width of the web, measured parallel to the axis of the shaft, shall not be less than 0·625 times the diameter required for the crank shaft, and the thickness of the metal round the eyes should be such that $h \times t^2$ is not less than 0·12 times $D^3$. That is:—

$$h \text{ should not be less than } 0{\cdot}625 \times D,$$
$$h \times t^2 \text{ should not be less than } 0{\cdot}12\ D^3,$$

where D is the diameter of the crank shaft, in inches;
*h* is the width of the crank web, in inches;
*t* is the thickness of metal round the eye measured radially, in inches.

15. Crank webs should be shrunk on the journals and crankpins, or be forced on by hydraulic pressure. They should be fitted with one or two keys or cylindrical dowels at the junction of the journals and webs. If only one dowel is fitted, its diameter should be 0·2 times the diameter of the shaft.

16. The liners to tail and tube shafts, if fitted, must be made of hard tough brass or bronze, shrunk on or forced on the shaft by hydraulic pressure, and no pins to secure them shall be fitted. Each liner must be cast in one piece; or, if made in two or more pieces, the junctions are to be made by electric welding or by fusing through the whole thickness of the liners. Joints made by butting, overlapping, stepping, or by soldering with soft solder are not permitted.

17. The thickness, in 64ths of an inch, of the liners of tail or tube shafts at the bearing portions should be not less than $13\sqrt{D}$, where D is the diameter of the shaft, in inches.

For the part of continuous liners between the bearing portions the thickness should be reduced to $10\sqrt{D}$.

18. The diameter of the screw for the nut at the end of the tail shaft should not be less than 0·7 times the diameter of the shaft under the liner. The depth of the nut should not be less than 0·625 the diameter of the shaft. The nut must be provided with efficient stops to prevent it from unscrewing.

19. The length of the aftermost bearing or stern bush shall not be less than four times the diameter required for the tail shaft.

20. The after end of the liner upon the tail shaft must be fitted watertight into the propellor boss.

21. Where hollow shafts are used, the diameter in inches of an equivalent solid shaft is $\sqrt[3]{\frac{D^4 - d^4}{D}}$, where D is the outside diameter and $d$ is the diameter of the hole in the hollow shaft, both in inches.

## Steam and Other Pipes.

**Section 17.**—1. No pipe made from the electro deposition of copper on a mandril shall be used for steam, feed delivery, blow-off, or scum.

2. All copper steam, feed, blow-off, and scum pipes must be properly annealed before fixing in place.

3. All copper pipes used for steam, feed, blow-off, and scum purposes subject to a pressure over 75 lbs. per square inch shall be solid drawn.

4. No steam pipe intended for a working pressure over 180 lbs. per square inch shall be of copper when the internal diameter exceeds 5 inches. No copper pipe shall be used for superheated steam.

5. All copper steam pipes on completion and prior to being fitted in place shall be subjected to an hydraulic test of at least twice the working pressure to which they will be subjected.

6. All copper feed delivery pipes shall be hydraulically tested to at least $2\frac{1}{2}$ times the working pressure of the boilers.

7. The working pressure to be allowed on copper pipes is to be determined by the following formula :—

$$WP = \frac{(t - 3) \times F}{D},$$

where WP is the working pressure, in lbs. per square inch ;
D is the internal diameter, in inches ;
$t$ is the thickness, in 100ths of an inch ;
F for solid-drawn steam pipes is 60 ;
for solid-drawn feed pipes is 48 ;
for brazed steam pipes 45 ;
for brazed feed pipes 36.

8. When copper pipes are bent, they must be made thicker than required by clause 7 to provide for the thinning at the bend. In no case should the radius of curvature at the centre line of the pipe be less than twice the external diameter of the pipe.

9. Steam and other pipes may be made of wrought iron or of wrought steel.

10. The process of welding the seams of iron or steel pipes shall be such that it is done by hammering or rolling the joint.

11. On completion of any work on iron or steel steam pipes which involves heating, whether for welding the joint, welding on flanges, bending the pipe, or for any other purposes, the pipe should be carefully annealed.

12. Mild steel for lap-welded or riveted steam pipes may have a tensile strength not exceeding 28 tons per square inch, with a minimum elongation of 25 per cent. on a standard test piece with a gauge length of 8 inches.

13. Feed pipes if made of steel should be solid drawn, cold finished.

14. All iron or steel steam pipes prior to being fitted in place shall be subjected to an hydraulic test of at least three times the working pressure to which they will be subjected.

15. All iron or steel feed delivery pipes shall be hydraulically tested to at least four times the working pressure of the boilers.

16. The working pressure to be allowed upon iron or steel steam and feed pipes shall be determined from the following formulæ :—

*a.* For solid-drawn cold finished steel steam pipes,

$$\mathrm{WP} = \frac{(t - 10)}{\mathrm{D}} \times 120\ ;$$

*b.* For solid-drawn hot finished steel steam pipes,

$$\mathrm{WP} = \frac{(t - 12)}{\mathrm{D}} \times 120\ ;$$

*c.* For lap-welded steam pipes of iron or steel whether with or without covering straps,

$$\mathrm{WP} = \frac{(t - 12)}{\mathrm{D}} \times 90\ ;$$

*d.* For feed delivery pipes,

$$\mathrm{WP} = \frac{(t - \cdot 8)}{\mathrm{D}} \times 100\ ;$$

where WP is the working pressure to which the steam pipes will be subjected, and in the case of feed delivery pipes is the boiler pressure ;

D is the internal diameter of the pipes, in inches ;

$t$ is the thickness, in 100ths of an inch.

## Boilers.

**Section 18.**—1. In the design and construction of marine boilers the following conditions should be observed :—

2. All steel plates, rivets, and bars used in the construction of boilers must be tested and found to conform to the requirements of the Rules (Sections 36, 37).

3. No steel plates subject to a direct tensile stress are to be welded except where the weld is covered by a butt strap or straps. For small steam domes, by special permission, where the welding is done by hammer, and the plates do not exceed $\frac{1}{2}$ inch in thickness, the straps may be omitted. The strength shall in such cases be assumed to be 50 per cent. of that of the solid plate.

4. All steel plates which are welded, dished, flanged, or locally heated are to be afterwards efficiently annealed.

5. Butt straps must be cut from plates and not from rolled strips.

6. All rivet holes must be drilled, and as far as possible they should be drilled in place. After drilling the plates the burrs should be removed and the faying surfaces of the plates cleaned, and the sharp outer edges of holes removed also.

7. No steel stays are to be welded. If plus threads are desired, the ends of the stay bars may be upset or the bars may be drawn down in the central portions from bars originally of the size of the ends. In either of these two cases the bars must be subsequently annealed throughout. In double-ended boilers the through longitudinal stays must be supported at or near the middle of their length.

8. Screw stays of combustion chambers when fitted with nuts should be, as far as possible, normal to the chamber plates. When this is not possible, they must be fitted with taper washers to provide a fair bed for the nuts.

9. Nuts to screw stays in combustion chambers shall be not less than $\frac{3}{4}$ inch thick for stays up to $1\frac{1}{2}$ inches diameter over threads, $\frac{7}{8}$ inch thick for $1\frac{5}{8}$ and $1\frac{3}{4}$ inch stays, 1 inch thick for $1\frac{7}{8}$ and 2 inch stays, and $1\frac{1}{8}$ inches thick for stays over 2 inches in diameter. The nuts shall be made of solid mild steel or of iron which must be without weld. The nuts for longitudinal stays shall be to the British standards appropriate to the diameters of the stays, the outside nuts having the thickness therein provided for ordinary nuts, and the inside nuts having the thickness provided for lock nuts.

10. All longitudinal stays and screw stays should have threads in accordance with British standard specification, and true to pitch, viz.:—

After June 30, 1921, all screw stays $1\frac{1}{4}$ inches in diameter and upwards should have 9 threads per inch, and all stays 2 inches in diameter and above passing through plates, and secured by nuts on each side of the plate, should have not more than 6 threads per inch.

It is desirable for the threads of all screw stays to be turned off between the parts fitting into the plates. It is also desirable that the outer ends of screw stays should have a hole $\frac{3}{16}$ inch diameter drilled axially to a distance $\frac{1}{2}$ inch beyond the inner face of shell and end plates.

11. When jointed longitudinal stays are fitted between the front and back tube plates it is desirable that they should be fitted with pins having an effective sectional area 25 per cent. in excess of that of the stay. The pins may be slack in the holes, the total slackness being not more than $\frac{1}{16}$ inch. The pins must be as close as possible to the shoulder of the eye forging. The shoulder of the forging should

have a diameter not less than 1 inch greater than the diameter of the hole.

**Section 19.**—1. No boiler, whether main or auxiliary, in which oil fuel is used, shall have a damper or other possible means of obstruction to draught in the uptake or funnel.

2. Manholes in cylindrical shells must have their shorter axes arranged longitudinally. Where the cylindrical shell is cut for a manhole, compensation must be provided and must be such that the strength in way of the hole is not less than that required for the longitudinal joint.

3. The end plates in the steam space in way of uptakes are to be shielded from contact with the heated gases.

**Section 20.**—1. When a flat plate is flanged to stiffen it at a manhole or sighthole, to permit the same working pressure as would be allowed upon an unpierced plate, the depth of the flange measured from the outer surface is to be at least equal to $\sqrt{t \times w}$, where $t$ is the thickness of the plate, in inches, and $w$ is the minor axis of the hole, in inches.

2. The doors to manholes, mudholes, and sightholes must be built up or pressed to shape and annealed, or made from one thickness of plate with a machined recess for the jointing material. Their spigot part or the recess must not have a greater clearance than $\frac{1}{16}$ inch all round, *i.e.* the axes must not be less than $\frac{1}{8}$ inch smaller than the holes in which they are fitted.

3. The studs for securing all doors should be screwed through the plate, and be fitted with nuts on the inside, or bolts may be used screwed through the plate with the heads inside.

4. All boilers should have, when possible, means for ingress to permit of examination and cleaning of the inner surfaces of plates and tubes exposed to flame. When the boilers are too small to permit of this, there must be sightholes and mudholes sufficiently large and numerous to permit of the inside being satisfactorily cleaned.

5. When the cross tubes of vertical boilers are large there must be a sighthole in the shell opposite to one end of each tube sufficiently large to permit the tube to be examined and cleaned. These sightholes must be in positions accessible for that purpose.

**Section 21.**—In all new boilers working at pressures up to 100 lbs. per square inch the hydraulic test must be twice the working pressure. For boilers working at pressures greater than 100 lbs. per square inch the hydraulic test pressure must be $1\frac{1}{2}$ times the working pressure plus 50 lbs. per square inch.

**Section 22.**—1. Every boiler must have at least two independent means of indicating the water level in it, and have marked on it in a contiguous position, easily seen, the level of the highest part of the combustion chamber. One of these means must be a glass gauge or an equivalent. The other may be a set of test cocks.

2. Test cocks, where practicable, should be fitted direct on the boiler shell. A set must consist of at least three cocks except in boilers 7 feet 6 inches in diameter and under, when there may be two.

For vertical boilers above 7 feet high the set of test cocks must consist of at least three.

3. All single-ended boilers over 16 feet mean diameter shall have a glass water gauge on each side. Single-ended boilers under 16 feet in diameter shall have one glass water gauge near one side and one set of test cocks near the other side.

4. All double-ended boilers shall have a glass water gauge near each end on opposite sides, and a set of test cocks at each end.

5. The cocks of all water gauges must be accessible from positions free from danger in the event of the glass breaking.

6. If the water gauges are not fitted directly to the shell of the boiler, but to stand pillars or columns, it is desirable that these pillars or columns should be bolted directly to the shell of the boiler. If they are connected to the boiler by means of pipes, the pipes must have terminal cocks, not valves, fitted direct to the boiler shell. For boilers exceeding 10 feet in diameter the pillars shall not be less than 2½ inches, and the connecting pipes not less than 1½ inches internal diameter. For boilers exceeding 7 feet 6 inches but not exceeding 10 feet in diameter, the pillars shall not be less than 2 inches, and the pipes not less than 1¼ inches internal diameter; and for boilers 7 feet 6 inches in diameter and under, the pillars shall not be less than 1¾ inches, and the pipes not less than 1 inch internal diameter. The upper ends of the connecting pipes must be arranged so that there is no pocket or bend where an accumulation of water from the condensation of the steam can lodge. They should not pass through the uptake if they can be otherwise arranged. If, however, this condition cannot be complied with, they may pass through it by means of a passage at least 2 inches clear of the pipe all round, open for ventilation.

7. A salinometer cock or valve must be fitted direct to each boiler in a convenient position. It must not be on the water gauge standpipe.

8. Each boiler shall have a separate steam-pressure gauge. Double-ended boilers shall have a pressure gauge at each end. The gauges are to be placed where they are easily seen.

9. Each boiler must have at least two independent means of feed, each with its own check valve. In vertical boilers one of the means of feed may be an injector.

**Section 23.**—1. Each boiler must have a blow-off valve fitted direct to the shell. The valve and its connections to the sea need not be more than 1½ inches in diameter, and may be generally about $\frac{1}{10}$ inch in diameter for each foot in diameter of the boiler. It should, however, be not less than ¾ inch in diameter.

2. Blow-off valves and scum valves (when these latter are fitted) of two or more boilers may be connected to one common discharge, but when thus arranged there must be screw-down non-return valves fitted for each boiler to prevent the possibility of the contents of one boiler passing to another.

3. The blow-off cock or valve on the ship's side must be fitted above the level of the stokehold plates, in an accessible position, and must

be arranged so that it can be readily seen whether it is open or shut. The cock handle must not be capable of being removed unless it is shut, and if a valve is fitted the wheel must be fixed to the spindle. The cock or valve must be fitted with a spigot passing through the plating and a brass or gun-metal ring on the outside.

**Section 24.**—One main stop valve must be fitted to each boiler direct on the shell of the boiler. There shall be as few auxiliary stop valves as possible so as to avoid piercing the boiler shell more than is absolutely necessary. The arrangement, however, must be such that when more than one boiler is fitted it is possible to supply the steam whistle, the steam steering gear, and the electric light machinery from at least two boilers.

**Section 25.**—1. At least two safety-valves must be fitted to each boiler. They must be arranged so that the springs and valves are cased in, that the valves cannot be overloaded when steam is up, that they can be lifted by easing gear, and turned round on their seats by hand, and in case of fracture of springs they cannot lift out of their seats. Easing gear must be arranged to lift all the safety-valves on a boiler together, and must be workable from some accessible place, free from steam danger.

2. Vertical boilers having more than 100 square feet of total heating surface must have two safety-valves each not less than 1·5 inches diameter; those having less than 100 square feet may have one valve not less than 2 inches diameter.

3. All the safety-valves of each boiler may be fitted in one chest, which must be separate from any other valve chest and must be connected direct to the boiler by a strong and stiff neck, the passage through which should have a cross-sectional area at least equal to one-half the aggregate area of the safety-valves in the chest. Each safety-valve chest shall have a means of draining it; the drain pipe shall lead to the bilge or tank clear of the boiler.

4. The minimum aggregate area of the safety-valves of the ordinary type in each boiler, whether coal-fired or oil-fired and whether working under natural, forced, or induced draught, shall be found by the following formula:—

$$\text{Aggregate area of safety-valves in square inches}$$
$$= \text{Total heating surface of boilers in square feet} \times \left(\frac{C}{p+15}\right),$$

where $p$ is the working pressure, in lbs. per square inch. C being 1·25 for boilers using coal, and 1·5 for those using oil fuel, and for all boilers with closed stokehold forced draught.

5. The waste-steam pipe and the passages leading to it should have a cross-sectional area not less in square inches than 0·01 times the total heating surface of the boiler in square feet, nor in any case should it be less than 1·1 times the combined areas of the safety-valves as given by the above rule.

6. All safety-valves must be set to the required pressure under

steam. During a test of 15 minutes with the stop valves closed and under full firing conditions the accumulation of pressure must not exceed 10 per cent. of the loaded pressure. During this test no more feed-water should be supplied than is necessary to maintain a safe-working water level.

**Section 26.**—1. All boiler mounting valves over $1\frac{1}{2}$ inches diameter must have outside screws, and all are to be arranged to be shut with a right-hand motion of the wheels, and must have means for clearly indicating whether they are open or shut. The covers must be secured by bolts or studs.

2. All cocks and valves connected to the boiler shall be such that it is seen without difficulty whether they are open or shut. When boiler mountings are secured by studs, the studs must have a full thread holding in the plate for a length of at least one diameter. If the stud hole penetrates the whole thickness of the plate, the stud must be screwed right through the plate and be fitted with a nut inside the boiler. Where bolts are used for securing mountings, they must be screwed right through the plate with their heads inside the boiler.

3. Where a superheater is fitted which can be shut off from the boiler it must have a separate safety-valve fitted with easing gear. The valve as regards construction must comply with the regulations for ordinary safety-valves, but the easing gear may be fitted to be workable from the stokehold only. The superheater must also be fitted with a drain cock or valve to free it from water when necessary.

4. All stop and safety-valve chests and steam-pipe fittings when subjected to saturated steam only may be made of cast iron, but such valve chests and steam-pipe fittings when subjected to steam of a temperature above 425 degrees Fahr., must be of cast steel or other approved material.

## Rules for Determining the Working Pressure to be allowed in New Boilers.

### *Cylindrical Shells.*

**Section 27.**—1. For the cylindrical shells of steel marine boilers the maximum working pressure (which is designated by WP, and is in lbs. per square inch) to be allowed shall be calculated from the following formulæ:—

Where the thickness of the shell plates does not exceed $1\frac{3}{4}$ inches,

$$\mathrm{WP} = \frac{(t-2) \times \mathrm{S} \times \mathrm{J}}{\mathrm{C} \times \mathrm{D}}.$$

Where the thickness of the shell plates exceeds $1\frac{3}{4}$ inches, and the longitudinal seams are made with double butt straps,

$$\mathrm{WP} = \frac{(t \times \mathrm{S} \times \mathrm{J})}{2{\cdot}85 \times \mathrm{D}}.$$

In the above formulæ

$t$ is the thickness of the shell plate, in 32nds of an inch ;
S is the minimum tensile strength of the steel shell plates, in tons per square inch ;
J is the percentage of strength of the longitudinal seams calculated by the methods described below ;
C is a coefficient, which is 2·75 when the longitudinal seams are made with double butt straps, 2·83 when the longitudinal seams are made with lap joints and are treble-riveted, 2·9 when they are made with lap joints and are double-riveted, and 3·3 when they are made with lap joints and are single-riveted ;
D is the inside diameter of the outer strake of plating of the cylindrical shell measured in inches.

2. The percentage of strength of a riveted joint (J) is found from the following formulæ (i.), (ii.), (iii.): (i.) and (ii.) are applicable to any type of joint, (iii.) is applicable only to that type of joint in which the number of rivets in the inner rows is double that in the outer row. The lowest value given by the application of these formulæ is to be taken as the percentage of strength of the joint.

(i.) Percentage of strength of plate at joint as compared with solid plate

$$= \frac{100(p-d)}{p} ;$$

(ii.) Percentage of strength of rivets as compared with solid plate

$$= \frac{100(S_2 \times a \times n \times C)}{S_1 \times p \times T} ;$$

(iii.) Percentage of combined strength of the plate at the inner row of rivet holes and of the rivets in the outer row

$$= \frac{100(p-2d)}{p} + \frac{100(S_2 \times a \times C)}{S_1 \times p \times T} ;$$

where $p$ = pitch of rivets at outer rows, in inches ;
$d$ = diameter of rivet holes, in inches ;
$a$ = sectional area of one rivet, in square inches ;
$n$ = number of rivets which are fitted in the pitch $p$ ;
T = thickness of plate, in inches ;
C = 1·0 for rivets in single shear as in lap joints ;
C = 1·875 for rivets in double shear as in double butt-strapped joints ;
$S_1$ = minimum tensile strength of plates, in tons per square inch ;
$S_2$ = shearing strength of rivets, which is taken generally to be 23 tons per square inch, and may be 85 per cent. of the minimum tensile strength of the rivet bars.

3. Where the longitudinal seams are fitted with double butt-strapped joints, the outer butt strap should have at least 0·625 of the strength of the plate, and should be of sufficient thickness to permit of efficient caulking of its outer edges. The inner butt strap should be $\frac{4}{32}$ inch thicker than this.

In cases where the number of rivets in the inner rows is double the number in the outer row, this will require the thickness of the outer strap to be

$$t_o = \frac{5 \times (p-d)}{8 \times (p-2d)} \times \mathrm{T},$$

and that of the inner strap to be at least

$$t_n = \frac{5 \times (p-d)}{8 \times (p-2d)} \times \mathrm{T} + \tfrac{4}{32} \text{ inch}.$$

4. In all cases the clear space between a rivet hole and the edge of a plate should not be less than the diameter of the rivet hole, *i.e.* the centre of the rivet hole should be at least $1\frac{1}{2}$ diameters distant from the edge of the plate.

In joints whether lapped or fitted with butt straps, in which there are more than one row of rivets and in which there is an equal number of rivets in each row, the distance between the rows of rivets should be not less than $0{\cdot}33p + 0{\cdot}67d$ with zigzag riveting, or $2d$ with chain riveting.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are chain riveted, the distance between outer rows and the next rows should be not less than $0{\cdot}33p + 0{\cdot}67d$ or $2d$, whichever is the greater, and the distance between the rows in which there are the full number of rivets should be not less than $2d$.

In joints in which the number of rivets in the outer rows is one-half of the number in each of the inner rows, and in which the inner rows are zigzag, the distance between the outer rows and the next rows should be not less than $0{\cdot}2p + 1{\cdot}15d$, and the distance between the rows in which there are the full number of rivets should be not less than $0{\cdot}165p + 0{\cdot}67d$.

In the above $p$ is the pitch of the rivets in the outer rows.

5. The maximum pitch of the rivets in the longitudinal joints of boiler shells is to be—

$$\text{Maximum pitch in inches} = \mathrm{C} \times \mathrm{T} + 1\tfrac{5}{8} \text{ inches},$$

where T is the thickness of the plate, in inches, and C is a coefficient as given in the following table:—

| Number of Rivets per Pitch. | Coefficients for Lap Joints. | Coefficients for Double Butt-strapped Joint. |
|---|---|---|
| 1 | 1·31 | 1·75 |
| 2 | 2·62 | 3·50 |
| 3 | 3·47 | 4·63 |
| 4 | 4·14 | 5·52 |
| 5 | ... | 6·00 |

6. If holes are cut in the cylindrical shells of boilers for fixing of mountings, the diameters of the holes being greater than $2\frac{1}{2}$ times the thickness of the shell plating plus $2\frac{3}{4}$ inches, compensation must be fitted as in the case of manholes. (See Section 19, clause 2.)

7. When more than three screw stays pierce the cylindrical shell in a horizontal line, if $d$ is their diameter and $p$ the pitch, $\frac{100(p-d)}{p}$ should not be less than the percentage of strength required for the shell longitudinal joints. If this is not possible, the stays must be arranged out of line with one another longitudinally.

8. The riveting of the seams joining the end plates to the cylindrical shell shall be not less than 42 per cent. of that of the solid shell plate. Where the shell plates exceed $\frac{5}{8}$ inch in thickness, the seams connecting the shell plates to the end plates are to be double-riveted. Where the shell plates exceed $\frac{1}{2}$ inch in thickness, the intermediate circumferential seams of double-ended boilers are to be at least double-riveted.

9. The circumferential seam at or near the middle of the length of single-ended boilers shall have a strength of joint not less than 60 per cent. of the solid plate. The inner circumferential seams of double-ended boilers shall have a strength of joint not less than 62 per cent. of the solid plate. In any case there shall be three rows of rivets when single-ended boilers have shell plates over $1\frac{3}{8}$ inch in thickness and when double-ended boilers have shell plates over $1\frac{3}{16}$ inch in thickness.

10. The circumferential seams of the shells of vertical boilers shall have a strength of not less than 42 per cent. of the solid plate. When these seams are not complete circles, and when the shell plates exceed $\frac{5}{8}$ inch in thickness, the riveting shall be double.

### *Furnaces.*

**Section 28.**—1. The working pressure to be allowed on corrugated furnaces is to be determined by the following formula:—

$$WP=\frac{C(t-1)}{D},$$

where D is the external diameter measured at the bottom of the corrugations, in inches;
$t$ is the thickness of the furnace plate, in 32nds of an inch, measured at the bottom of the corrugation or camber;
C is a coefficient which is 480 for the Fox, Morison, Deighton, Purves, and other similar furnaces, and is 510 for the Leeds Forge Bulb Suspension furnace.

2. The working pressure to be allowed on plain furnaces or furnaces strengthened by the Adamson or other joints, and on the cylindrical bottoms of combustion chambers, is to be determined by the following formulæ, the least pressure obtained by either formula being taken:—

$$\mathrm{WP} = \frac{C(t-1)^2}{(L+24)\times D}$$

or

$$\mathrm{WP} = \frac{C_1}{D} \times [10(t-1) - L],$$

where D is the external diameter of the furnace or combustion chamber, in inches;
$t$ is the thickness of the furnace plate, in 32nds of an inch;
L is the length of the furnace or of combustion chamber bottom or the length between points of substantial support, in inches, measured from the centres of rivet rows or from the commencement of flange curvature, whichever is applicable;
C is 1450 where the longitudinal seams are welded, and 1300 where they are riveted;
$C_1$ is 50 where the longitudinal seams are welded, and 45 where they are riveted;
WP is the working pressure, in lbs. per square inch.

3. When the furnaces are tapered, the diameter to be taken for calculation purposes shall be the mean of that at the top and of that at the bottom where it meets the substantial support from flange or ring. The length for the same purpose shall be the distance from the centre of the row of rivets connecting the crown to the body of the furnace to the substantial support at the bottom of the furnace, or to a row of screwed stays connecting the furnace to the shell, provided the pitch of stays at the furnace does not exceed 14 times the thickness of the furnace plate when the stays are riveted at their ends, and 16 times when they are fitted with nuts. Such screwed stays must be in diameter over the threads not less than 2·25 times the thickness of the furnace plate.

4. Where the furnaces are spherical in form and convex upwards at their tops, and are without support from stays of any kind,

$$\mathrm{WP} = \frac{275(t-1)}{R},$$

where $t$ is the thickness of the top plate, in 32nds of an inch;
R is the outer radius of curvature of the furnace, in inches.

5. For the ogee ring which connects the bottom of the furnace to the shell, and sustains the whole load on the furnace vertically,

$$\mathrm{WP} = \frac{140(t-1)^2}{\mathrm{D} \times (\mathrm{D}-d)},$$

where $t$ is the thickness of the ogee ring, in 32nds of an inch;
D is the inside diameter of the boiler shell, in inches;
$d$ is the outside diameter of the lower part of the furnace where it joins the ogee line.

6. No furnace, plain or corrugated, should exceed $\frac{13}{16}$ inch in thickness.

*Flat Plates supported by Stays secured in Various Ways.*

**Section 29.**—1. The working pressure to be allowed on flat plates supported by stays is to be calculated by the following formula:—

$$\mathrm{WP} = \frac{(t-1)^2 \times \mathrm{C}}{a^2+b^2}.$$

In this formula and in that in paragraphs 8 to 13,
WP is the working pressure, in lbs. per square inch;
$t$ is the thickness of the flat plate, in 32nds of an inch;
$t_w$ is the thickness of the washers, strips, or doublings employed, also in 32nds of an inch;
$a$ is the distance apart of the rows of stays, in inches;
$b$ is the pitch of the stays in the rows, in inches;
C is a coefficient which varies with the method of fixing the stays as follows:—

Where the plates are exposed to flame and the stays are screwed into the plate and the ends are riveted over, C = 50. Where the plates are not exposed to flame and the stays are screwed into the plate and their ends are riveted over, C = 57. In these cases the thickness of the plate must be at least half the diameter of the stay required by the Rules.

Where stay tubes are screwed into tube plates and expanded, C = 52. If they are fitted with nuts, C = 72.

Where the plates are exposed to flame and the stays are screwed into the plate and fitted with nuts on the outside, C = 75. Where the plates are not exposed to flame, C = 86.

Where the stays pass through the plates not exposed to flame and are fitted with nuts inside and outside, C = 96.

2. Where plates are stiffened by flanging, the inner radius of which is not greater than 2½ times the thickness of the plate, for the support thus given, C = 110 when the plates are not exposed to flame, and C = 96 when they are exposed to flame. The pitch is to be reckoned from the commencement of the curvature.

3. For portions of plate where the stays are irregularly pitched, $d^2$ is to be used instead of $a^2+b^2$, $d$ being the diameter of the largest circle

which can be drawn passing through not less than three points of support—viz. the centres of stays or the commencement of the curvature of flanging, whichever is applicable. In this case C is to be taken as the mean of the values appropriate for the points of support.

4. For the tops and sides of combustion chambers the distance between the rows of stays nearest to the back tube plate or the back plate respectively and the commencement of curvature of these plates at their flanges shall not be greater than $a$.

5. It is desirable that the stays of the combustion chambers should be so placed that the seams of the plates can be caulked without removing the stay nuts.

6. For the tops of combustion chambers where they are joined to the sides by curved portions, if the outer radius of the curved portion is less than half the allowable distance between the girders, the distance between the first girder and the inner surface of the side plate should not exceed the allowable distance between the girders. If the radius of the curved portion is greater than half the allowable distance between the girders, the width of the flat portion measured from the centre of the girder should not be more than half the allowable distance between the girders.

7. Where portions of plate are supported by stays secured in different ways, the value of C to be taken is the mean of the values appropriate to the method of securing the supporting stays.

8. Where the plates are supported by stays passing through them and are fitted with nuts inside and washers and nuts outside, the diameter of the washers being at least 3·5 times that of the stay, and their thickness at least two-thirds of that of the plate, but not greater than that of the plate, the working pressure may be

$$\mathrm{WP} = \frac{100}{a^2+b^2}[(t-1)^2 + 0{\cdot}15 t_w{}^2].$$

9. Where the washers have a diameter of at least two-thirds of the pitch of the stays and a thickness of at least two-thirds of the thickness of the plate, but not greater than that of the plate, and are riveted to the plate in an efficient manner :—

$$\mathrm{WP} = \frac{100}{a^2+b^2}[(t-1)^2 + 0{\cdot}35 t_w{}^2].$$

10. Where the plate is stiffened by strips at least two-thirds of the pitch of the stays in breadth and have a thickness at least two-thirds of that of the plate, but not greater than that of the plate, and are riveted to the plate in an efficient manner :—

$$\mathrm{WP} = \frac{100}{a^2+b^2}[(t-1)^2 + 0{\cdot}55 t_w{}^2].$$

11. Where the plates are fitted with doubling plates having a

thickness of at least two-thirds of that of the plate, but not greater than that of the plate, and are riveted to them :—

$$WP = \frac{100}{a^2 + b^2}[(t-1)^2 + 0{\cdot}85 t_w^2].$$

12. For the portions of tube plates in the nests of tubes,

$$WP = \frac{C(t-1)^2}{p^2},$$

where $t$ is the thickness of the tube plate, in 32nds of an inch ;
 $p$ is the mean pitch of the stay tubes supporting any portions of the plate (being the sum of the four sides of the quadrilateral divided by 4) ;
 $C = 38$ when the stay tubes are screwed and expanded into the plate and no nuts are fitted ;
 $C = 49$ when the stay tubes are screwed and expanded into the plates and fitted with nuts.

13. For the wide water spaces of front tube plates between the nests of tubes and between the wing rows of tubes and the shell,

$$WP = \frac{C[(t-1)^2 + 0{\cdot}55 t_w^2]}{a^2 + b^2},$$

where $t$ is the thickness of the front tube plate, in 32nds of an inch ;
 $t_w$ is the thickness of the doubling plate, when so fitted, in 32nds ;
 $a$ is the horizontal pitch of stay tubes, in inches, measured across the wide water space ;
 $b$ is the vertical pitch of stay tubes in the bounding rows, in inches, measured from centre to centre ;
 $C = 52$ when the stay tubes are screwed and expanded into the tube plates, and no nuts are fitted ;
 $C = 72$ when the stay tubes are screwed and expanded into the tube plates, and nuts are fitted to each stay tube ;
 $C = 63$ when the stay tubes are screwed and expanded into the tube plates, and nuts are fitted only to alternate stay tubes.

14. If steel of less strength than 26 tons per square inch is used for flat plates, the working pressure allowed shall be correspondingly reduced.

**Section 30.**—The pressure to be allowed on tube plates shall be calculated by the following formulâ, in which the compressive stress is taken at 14,000 lbs. per square inch :—

$$WP = 875 \times \frac{(D-d) \times t}{W \times D},$$

where $t$ is the thickness of the tube plate, in 32nds inch ;
D is the horizontal distance apart of the tubes, centre to centre, in inches ;
$d$ is the internal diameter of the plain tubes ;
W is the width of combustion chamber measured inside from tube plate to back chamber plate, or between tube plates in double-ended boilers with combustion chambers common to two opposite furnaces.

**Section 31.**—When vertical boilers have a nest or nests of horizontal tubes so that there is direct tension on the tube plates due to the vertical load on the boiler ends or to their acting as horizontal ties across the shell, the thickness of the tube plates and the spacing of the tubes must be such that the section of metal taking the load is sufficient to keep the stress within that allowed on the shell plates.

Further, each alternate tube in the outer vertical rows of tubes must be a stay tube. The tube plates between the stay tubes must be in accordance with the rules for tube plates as in Section 29, clause 12, and, in addition,

$$\mathrm{WP} = \frac{(t-2) \times \mathrm{S} \times (p-d) \times 100}{2{\cdot}9 \times \mathrm{D} \times p},$$

where S is the minimum tensile strength of the steel plate, in tons per square inch ;
$t$ is the thickness of the tube plate, in 32nds of an inch ;
D is twice the radial distance of the centre of the outer row of tube holes from the axis of the shell, in inches ;
$p$ is the vertical pitch of tubes ;
$d$ is the diameter of the tube holes, in inches.

### *Girders.*

**Section 32.**—For girders supporting the tops of combustion chambers the following formula is to be used :—

$$\mathrm{WP} = \frac{\mathrm{C} \times d^2 \times t}{(\mathrm{L}-\mathrm{P}) \times \mathrm{D} \times \mathrm{L}} \times \frac{\mathrm{S}}{28},$$

where $d$ is the depth of the girder at centres, in inches ;
$t$ is the thickness of the girder at centre, when this is a forging, or the sum of the thicknesses of the plates when the girder is made of two plates, measured in 32nds inch ;
L is the length of the girder in inches, measured between the tube plate and back chamber plate inside, or between tube plates in chambers common to two opposite furnaces ;
P is the pitch of stays supported by the girder, in inches ;
D is the distance apart of the girders, centre to centre, in inches ;

S is the minimum tensile strength of the steel plates forming the girder, in tons per square inch. In the case of forged girders S is to be taken as 24 for iron and 28 for steel;
C is a coefficient as follows:—

$$C = \frac{n}{n+1} \times 495,$$

when the number of stays in each girder is odd;

$$= \frac{n+1}{n+2} \times 495,$$

when the number of stays in each girder is even, $n$ being the number of stays to each girder.

## *Stays.*

**Section 33.**—1. For screw stays with threads not coarser than 9 threads per inch, made of steel or of special wrought iron tested to the requirements of the Rules (Sections 36, 37), the following formula is to be used, but in no case must the stress exceed 9000 lbs. per square inch:—

$$WP = \frac{(d - 0{\cdot}267)^2 \times 8250}{a},$$

where $d$ is the diameter of the stay over the thread, in inches;
$a$ is the area, in square inches, supported by one stay.

2. For steel longitudinal stays with threads not coarser than 6 threads per inch the working pressure is to be calculated from the following formula, but in no case must the stress exceed 11,000 lbs. per square inch when steel of a minimum tensile strength of 28 tons per square inch is used:—

$$WP = \frac{(d - 0{\cdot}340)^2 \times 9500}{a} \times \frac{S}{28},$$

where $d$ is the diameter of the stay over the thread, in inches;
$a$ is the area, in square inches, supported by one stay;
S is the minimum tensile strength of the steel, in tons per square inch.

3. In cases where longitudinal stays are made with enlarged ends and the body of the stay is smaller in diameter than at the bottom of the thread, and in cases where coarser threads than 6 per inch are used, the working pressure is to be calculated from the following formula:—

$$WP = \frac{(d^1 - 0{\cdot}125)^2 \times 9500}{a} \times \frac{S}{28},$$

where $d^1$ is the diameter of the stay at the bottom of the thread or at the smallest part of the body.

*Boiler Tubes, Plain and Stay*

**Section 34.**—1. The following table gives the working pressures permissible with plain boiler tubes of standard thicknesses, whether of lapwelded wrought iron or of lapwelded or seamless mild steel:—

| Outside Diameter of Tube in Inches. | Standard Thicknesses in L.S.G. | | | | Working Pressures in lbs. per sq. inch. | | | |
|---|---|---|---|---|---|---|---|---|
| | A | B | C | D | A | B | C | D |
| 2 | | 11 | 10 | 9 | | 155 | 215 | 300 |
| $2\frac{1}{4}$ | 11 | 10 | 9 | 8 | 140 | 190 | 260 | 315 |
| $2\frac{1}{2}$ | 11 | 10 | 9 | 8 | 125 | 175 | 230 | 300 |
| $2\frac{3}{4}$ | 11 | 10 | 9 | 8 | 110 | 160 | 215 | 275 |
| 3 | 10 | 9 | 8 | 7 | 140 | 190 | 250 | 300 |
| $3\frac{1}{4}$ | 10 | 9 | 8 | 7 | 130 | 180 | 230 | 280 |
| $3\frac{1}{2}$ | 10 | 9 | 8 | 7 | 120 | 165 | 215 | 260 |

2. On stay tubes, whether of wrought iron or of lapwelded steel, a working stress of 7500 lbs. per square inch of the net sectional area at the bottom of the thread is allowed.

3. The minimum thickness of stay tubes measured under the threads shall be $\frac{1}{4}$ inch for marginal stay tubes and $\frac{3}{16}$ inch for other stay tubes.

4. Stay tubes are to be screwed at both ends with continuous threads, and the holes in the tube plates are to be tapped with continuous threads. The thread should not be finer than 10 threads per inch. It is desirable, however, that they should be screwed to the standard 9 threads per inch, and after June 30, 1921, this should be the rule. The stay tubes are to be expanded by roller expanders and not made tight by caulking only.

5. No nuts are to be fitted to stay tubes at the combustion chamber end.

6. If stay tubes are required to have their thickness increased at the screwed ends so that the thickness at the bottom of thread is practically the same as in the body of the tube, the thickening is to be attained by upsetting and not by any welding process, and the tubes are to be annealed after the upsetting.

**Section 35.**—1. For ends of steam chests, crowns of vertical boilers, etc., dished outwards to partial spherical form and not fitted with stays, the following formula is to be used:—

$$\mathrm{WP} = \frac{15 \times \mathrm{S}(t-1)}{\mathrm{R}},$$

WP is the working pressure, in lbs. per square inch ;
$t$ the thickness, in 32nds of an inch ;
R the inner radius of curvature of the end in inches, which shall not exceed the diameter of shell ;
S the minimum tensile strength of plates.

The inside radius of curvature at the flange connecting the end to the cylindrical shell must not be less than four times the thickness of the end plate, and in no case less than 2·5 inches.

2. When the end has a manhole in it, $\frac{4}{32}$ inch must be added to the thickness of plate.

If the plate at the manhole is stiffened by flanging, the total depth of the flange from the outer surface, in inches, is to be at least

$$=\sqrt{t \times w},$$

$t$ is the thickness of plate, in inches,
$w$ is the minor axis of the hole, in inches.

3. When the end or crown is a complete hemisphere without stays or other supports, and is made in more than one plate, the working pressure to be allowed is given by the following formula :—

$$\mathrm{WP}=\frac{(t-2)\mathrm{S} \times \mathrm{J}}{\mathrm{C} \times \mathrm{R}},$$

where $t$ is the thickness of the plates, in 32nds of an inch ;
S is the minimum tensile strength of the steel plates, in tons per square inch ;
J is the strength of riveted joint per cent. of the solid plate ;
R is the inner radius of curvature, in inches ;
C is a coefficient which for treble riveting is 2·83, for double riveting is 2·9, and for single riveting is 3·3.

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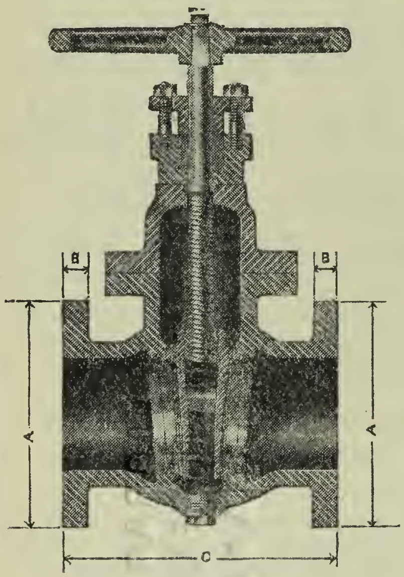
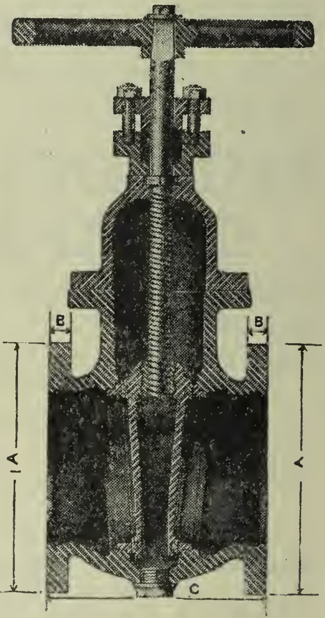
# "Sentinel" Oil Valves

H.P. For Pressures to 250 lbs & Over



L.P. For Pressures to 125 lbs.



| Size of Valve | 2″ | 2½″ | 3″ | 3½″ | 4″ | 5″ | 6″ | 7″ | 8″ | 9″ | 10″ | 12″ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| A | 6½″ | 7¼″ | 8″ | 8½″ | 9″ | 11″ | 12″ | 13¼″ | 14½″ | 16″ | 17″ | 19¼″ |
| B | ⅞″ | ⅞″ | 1″ | 1″ | 1⅛″ | 1¼″ | 1¼″ | 1⅜″ | 1⅜″ | 1½″ | 1⅝″ | 1¾″ |
| C | 10″ | 10½″ | 11¼″ | 12″ | 13″ | 13¾″ | 14½″ | 15¾″ | 14″ | 15″ | 16¼″ | 17½″ |

| Size of Valve | 2″ | 2½″ | 3″ | 3½″ | 4″ | 5″ | 6″ | 7″ | 8″ | 9″ | 10″ | 12″ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| A | 6″ | 6½″ | 7½″ | 8″ | 8½″ | 10″ | 11″ | 12″ | 13¼″ | 14½″ | 16″ | 18″ |
| B | ¾″ | ¾″ | ¾″ | ¾″ | ⅞″ | ⅞″ | ⅞″ | 1″ | 1″ | 1″ | 1″ | 1⅛″ |
| C | 7″ | 7½″ | 8″ | 8½″ | 9″ | 10″ | 10½″ | 11″ | 11½″ | 12″ | 13″ | 14″ |

CATALOGUE 25 S. GIVING FULL PARTICULARS OF ABOVE & MANY OTHER TYPES OF SENTINEL VALVES FREE ON APPLICATION.

## Alley & MacLellan, Ltd.

## Sentinel Works, GLASGOW.

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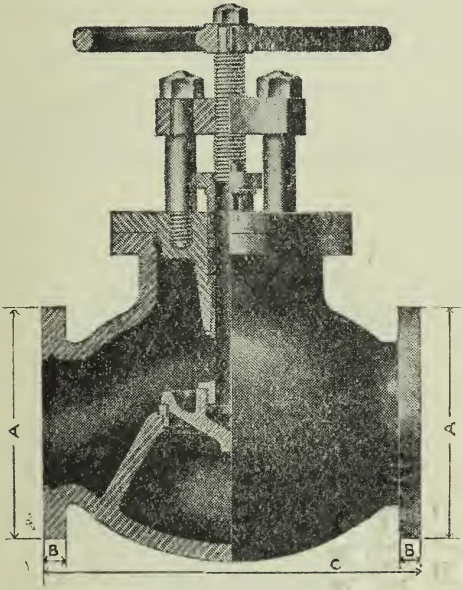

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- VALVE FREE TO ROTATE.
- NICKEL ALLOY SPINDLE, AVOIDS EXPANSION TROUBLES.
- ALL PARTS TO GAUGE AND INTERCHANGEABLE.

DIMENSIONS.

| Bore of Valve | 2" | 2½" | 3" | 3½" | 4" | 5" | 6" | 7" | 8" | 9" | 10" |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Dimension A .. | 6½" | 7½" | 8" | 8½" | 9" | 11" | 12" | 13¼" | 14½" | 16" | 17" |
| „ B .. | ⅞" | ⅞" | 1" | 1" | 1⅛" | 1⅛" | 1¼" | 1⅜" | 1⅜" | 1½" | 1½" |
| „ C .. | 10" | 11" | 12" | 13" | 14" | 17" | 20" | 22" | 24" | 26" | 28" |

CATALOGUE 25 S. GIVING FULL PARTICULARS OF ABOVE & MANY OTHER TYPES OF SENTINEL VALVES FREE ON APPLICATION.

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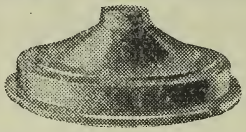

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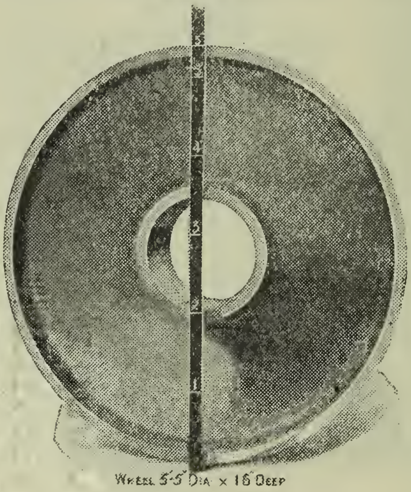

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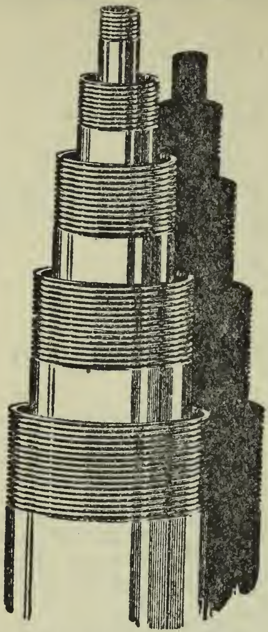

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